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NATIONAL BUREAU OF STANDARDS REPORT

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ANALYSIS FOR THE DETERMINATION OF AN AIR CONDITIONING CRITERIA

Report to

The Department of Housing and Urban Development

U.S. DEPARTMENT OF COMMERCE

NATIONAL BUREAU OF STANDARDS

NATIONAL BUREAU OF STANDARDS

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ANALYSIS FOR THE DETERMINATION AN AIR CONDITIONING CRITERIA

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Executive Summary

Presented in this report is detailed information describing the need for a technical base for air conditioning criteria. The report includes results of a feasibility study for evaluating indoor habitability conditions for air conditioned as well as for non-air conditioned buildings during summer months. In the feasibility study the indoor habitability condition is determined by calculation using a computerized simulation technique which follows a detailed heat transfer analysis of building-human body systems. A concept of "Predicted Indoor Habitability Index" (PIHI) is introduced to designate an index for describing the indoor thermal environment. PIHI is designed as an index that relates the calculated indoor temperature, humidity, air velocity, and mean radiant temperature to the physiological as well as psychological response of occupants in such a manner that the short term as well as long term exposure in the non-air conditioned buildings can be evaluated. There are several published types of physiological indices (both comfort and discomfort). A study of these indices with respect to their applicability to the PIHI concept is given.

In the introduction the need for an air conditioning criteria is given together with the manner in which it can be established. This involves a detailed survey of building data and weather data as well as a precise thermal simulation of buildings. The second part of the report describes the basic mathematical relationships used to predict indoor temperature and humidity as a response of the building to climatic factors. The third part presents progress to date on compiling and analyzing building data. This effort is essential in determining air conditioning criteria and entails the development of heat transfer characteristics, air leakage characteristics, and thermal mass characteristics which play very important roles in determining the indoor conditions of any building unit. An analysis is done to find the range and distribution of these basic parameters so that a meaningful building classification useful for this approach can be made. The third part will be the subject of a separate NBS numbered report.

In Part 4, a review is given of various physiological indices currently available. It has been possible to include very recent information in this part. Part 5 gives the details of a comprehensive feasibility study of a thermal simulation made on an apartment building located both in Jersey City, New Jersey and Macon, Georgia. Algorithms for many of the physiological indices were used in conjunction with the thermal simulation to give hour by hour values of these indices in the apartment. An examination of the results of this study reveals the extent and duration of undesirable indoor conditions when this apartment is not air conditioned. The magnitude of computational effort appears formidable if detailed

simulation calculations are to be carried out for the many combinations of building parameters and climate zones in the United States. Therefore, an attempt was made to obtain statistical correlations between the indoor and outdoor conditions for this apartment. The attempt was reasonably successful and a short-cut technique for establishing the criteria is indicated.

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1. Introduction

At the present time in the United States government agencies, the decision to air condition a single family or multifamily dwelling is usually based on cost and general location of the housing unit (i.e., northern or southern part of the country). The United States Air Force, for example, specifies in their Manual 88-8 that family housing located in a region where the wet-bulb temperature is above 67 °F for 1000 hours or more during the hottest six consecutive months of the year should have air conditioning. The General Services Administration uses the degree-day concept in specifying the guidelines for air conditioning their cars. If there are more than 700 cooling-degree-days based on 65 °F in a year (at that particular location), then air conditioning should be installed. In addition, the Office of Management and Budget gives a criteria similar to that of the Air Force in relation to the air conditioning of federal buildings. For lack of additional information on family housing, the Air Force criteria was adopted by the Department of Housing and Urban Development and is present in their four volumes "Guide Criteria for the Design and Evaluation of Operation Breakthrough". Using this as a basis for the decision on air conditioning is a substantial improvement over no criteria at all; however, it is felt that other factors besides weather should be considered.

One of the most important items that govern the indoor environment is the "thermal performance" of the enclosing structure. J. F. van Straaten of the Building Research Institute of South Africa discusses in his recent book Thermal Performance of Buildings (1) the way different structures respond when subjected to periodically varying solar conditions and environmental temperatures. The overall heat transfer coefficient for a wall, which gives a direct indication of the resistance to heat flow under steady-state conditions, is not the sole indicator of the heat flow to the inside environment when unsteady-flow conditions exist. The mass of the wall, floor and ceiling, which governs the energy storage capacity, is also a pertinent parameter to be considered.

Van Straaten reports the results of an experimental study where small 4 room single storey structures of about 500 ft^2 floor area were built, not air conditioned, unoccupied, and instrumented to observe the indoor environment as a function of time. Under warm arid conditions the massive brick buildings proved to be superior in minimizing the daily variation of the indoor temperature. The reason is that the "time lag" associated with the massive structure was such that the indoor environment remained sufficiently cool during the hot part of the day (79 °F compared with 84 °F outdoor temperature at noon) and also remained sufficiently warm during the night (74 °F compared with 61 °F outdoor temperature at 5:00 a.m.). The light timber structures with essentially no "time lag" allowed the indoor temperature to increase too much during the day (93 °F compared with 87 °F outdoor temperature at 2:00 p.m.) and to become too cool during the night (62 °F compared with 61 °F outdoor temperature at 5:00 a.m.).

On the other hand, van Straaten concludes from experiments conducted in Durban, South Africa where the climate is warm and humid and the daily outdoor temperature variation is considerably less even though the daily average temperature is approximately the same and solar radiation intensities are high, that massive buildings were a distinct disadvantage. The walls could not cool down sufficiently during the night to provide reasonably comfortable temperatures in the enclosures during the day. Other interesting results of van Straaten's research include the improvement in performance of the lightweight structures in warm and dry climates by the addition of concrete floors and by painting the exterior of the structure white. It should be apparent from this type of investigation that climate as well as building construction plays a significant role in determining a need for air conditioning.

Obviously, another factor to be considered in establishing the air conditioning criteria is the range of indoor thermal conditions under which people remain comfortable or not overly uncomfortable. The American Society of Heating, Refrigerating and Air-Conditioning Engineers Standard 55-66 is very specific in its stipulation of thermal comfort conditions. In a region between 3 inches above the floor and 72 inches above the floor and at least 2 feet from any wall, the following should be met at all times.

1. The dry bulb temperature shall be between 73 and 77 °F.
2. The relative humidity shall not exceed 60 percent and be greater than 20 percent.
3. The air motion shall not exceed 45 fpm and be greater than 10 fpm.

In other words, if the indoor thermal environment of non-air conditioned buildings falls into this condition, it is obvious that the building needs no air conditioning even if it is located in the area where the climatic criteria indicates the need for it. Because of the inflexibility of the statement "at all times", it may not be appropriate to incorporate this criteria into the air conditioning design criteria. A better evaluation of the tolerance of people to varying indoor conditions above and below the recommended steady state comfort conditions, magnitude and frequency is needed.

There are also other basic considerations that could be used in establishing an air conditioning criteria. For example, one might be able to maintain reasonable conditions in a space by a large amount of forced or natural ventilation yet it is possible that it would not be feasible due to air and/or noise pollution problems. It might be cheaper to design completely closed-in structures with air conditioning than more standard type buildings without air conditioning. Considerations such as these will not be dealt with initially in this study.

The manner in which the criteria, based on thermal performance alone, could be established is depicted in Figure 1. All combinations of selected weather data and selected building data are combined in the thermal computer simulation, the output of which is the space conditions. The frequency and duration of the various indoor parameters are then compared with the PIHI to determine whether the space should be conditioned or not. The studies completed to date, discussed in Part 5, show the feasibility of such an approach.

It should also be noted that the concept of determining a need for air conditioning based on building thermal performance is not new. Givoni in his latest book (2) introduces a "Building Bioclimatic Chart". It is essentially a psychrometric chart on which he has drawn an enclosed region indicating thermal comfort conditions. From experience and study, three additional regions have been drawn: one to indicate the range of outdoor conditions where choice of building materials and design features (large thermal resistance, large thermal mass, etc.) make comfort conditions inside attainable without air conditioning; a second region to indicate the outdoor conditions where efficient ventilation through the structure could result in comfort conditions; and a last region showing the range of outdoor conditions where mechanical cooling would be required.

In a recent article of the ASHRAE Journal (3), Nevins and McNall of Kansas State University discuss a proposed method of classifying air conditioning systems. They suggest that the environment of a given occupied space can be evaluated in terms of the human response to that environment and that an appropriate classification or "grade" can be assigned to the system serving that space based upon the functional performance in the space. Their ideas are so much in line with the kind of analysis presented in this report that their concluding two paragraphs of "Suggested Future Action" are included here:

> "We suggest that a committee be formed to examine the various methods of establishing a "percent dissatisfied" or other functional rating method for an environment which seems most appropriate in the light of all present data. Such a rating

method must take into account practical means of measuring the environmental variables.

Any functional specifications must then be met with practical equipment. Other committees must assess the functional performance of systems of various types. The system in this case includes not only the air-conditioning equipment and its controls but also the building, its construction details and the weather conditions in which the system operates. This difficult problem must be met by field experiments, that establish for various air-conditioning equipment and control combinations, which of the construction details (window sizes, single or double, etc. and wall "U" values) and what types of climates (temperature, wind, solar exposure, etc.) provide what level of thermal environment in the space to be compared with the corresponding functional "percent dissatisfied" or other criteria. When all this is accomplished, each air-conditioning system and controls combination could be tabulated with ranges of climate and construction details which provide various classifications of thermal (and economical performance."

Even though they are discussing a different end product, the all-encompassing analysis suggested is very similar to that presented in this report.

2. Calculation of Space Conditions by Building Thermal Simulation on a Digital Computer

Numerous papers have been published and calculation manuals prepared in the area of heating/cooling load determination. Very few of these, however, have dealt with the indoor temperature estimates for the condition where air conditioning is non-existent or inadequate. The room temperature prediction is, on the other hand, very actively studied in Europe, Japan, Israel, and India where mechanical cooling is still very expensive for the majority of dwellings and consequently the design of a house to make the indoor environment habitable is extremely important.

The heat transfer calculations for room temperature prediction are similar to the heating/cooling load calculations; the former are somewhat more complex than the latter because they require exact heat balance calculations for the room air, surrounding walls and infiltrating outdoor air. Since the major importance is the hour by hour room temperature profile, it is necessary to account for the transient heat conduction and storage of building mass as affected by the climatic parameters.

In order to determine temperature and humidity of non-air conditioned rooms responding to randomly fluctuating outdoor climatic conditions, a computer program to simulate hourly performance of building heat gain and heat storage has been developed. The program consists of various subroutines for calculating heat gains, which are similar to those recommended by the ASHRAE Task Group on Energy Requirements (4). The detail of these subroutines are, therefore, not given in this report. One major extension of the program beyond the recommended ASHRAE TG subroutine is the routine called RMTMP, which solves for room air temperature. General mathematical expressions of this routine are described as follows: A general expression for the heat exchange at an interior surface of the room is

$$\frac{QK_{i,t}}{A_i} = HC_i \; (T_{a,t} - T_{i,t}) + \sum_{k=1}^{M} HD_{ki} \; (T_{k,t} - T_{i,t}) + RS_{i,t} + RL_{i,t} + RE_{i,t} \tag{1}$$

for $i = 1, 2, \ldots M$

where $QK_{i,t}$ = rate of heat conducted into surface i at the interior surface at time t

A_i = cross sectional area of the surface

HC_i = convective heat transfer coefficient at interior surface i

HD_{ki} = radiation heat transfer factor between interior surfaces i and interior surface k

$T_{a,t}$ = room air temperature at time t

$T_{i,t}$ = uniform temperature of surface i at time t

$T_{k,t}$ = uniform temperature of surface k at time t

$RS_{i,t}$ = rate of solar heat coming through window and absorbed by surface i at time t

$RL_{i,t}$ = rate of heat radiated from lights and absorbed by surface i at time t

$RE_{i,t}$ = rate of heat radiated from appliances and occupants and absorbed by surface i at time t

The total number of interior surfaces to be considered for this general expression is designated by M and it varies depending upon the complexity of the room construction. If, for example, each of four walls are made of four different segments of different construction and if each segment of the wall contains a window and a door, M will be 2 x 4 x 4 + 2 = 34 with the assumption that ceiling and floor are of uniform structure. The calculation of the heat conduction $QK_{i,t}$ is performed by the response factor technique for heavy construction, such as exterior walls, roof and floor and by the use of overall heat transfer coefficient U (steady state calculation) for doors and windows (lightweight construction).

The value of $QK_{i,t}$ depends upon the exterior surface temperature $T_{o,t}$ at the opposite side of the given interior surface by the following expression

$$\frac{QK_{i,t}}{A_i} = \sum_{j=o}^{\infty} X_{i,j}\, T_{i,t-j} - \sum_{j=o}^{\infty} Y_{i,j}\, T_{o,t-j} \qquad (2)$$

where $X_{i,j}$ and $Y_{i,j}$ ($j = 0, 1, \ldots \infty$) are the response factors which are to be determined as functions of wall (roof) construction. The method to calculate the response factors is given in reference (5).

On the other hand, the following expression describes the heat transfer process at the exterior of the surface:

$$\frac{QK_{o,t}}{A_i} = HC_o\ (T_{o,t} - DB_t) + \sum_{\ell=1}^{M'} HD_{o,\ell}\ (T_{o,t} - T_{\ell,t}) - RS_{o,t} \qquad (3)$$

where $QK_{o,t}$ = rate of heat conducted outward at the exterior side at time t

HC_o = convection heat transfer coefficient at the exterior side, which is a function of wind velocity

$HD_{o,\ell}$ = radiation heat transfer factor between the exterior side and other surface ℓ, which sees the surface. Sky and ground may be treated as one of these ℓ surfaces

$RS_{o,t}$ = rate of solar radiation absorbed at the exterior side at time t

$T_{o,t}$ = temperature of the exterior surface at time t

DB_t = outdoor air dry-bulb temperature at time t

Again in Equation (3) the conducted heat rate $QK_{o,t}$ is calculated by the response factors for the heavyweight structures such as walls and roofs by

$$\frac{QK_{o,t}}{A_i} = \sum_{j=o}^{\infty} Y_{i,j}\ T_{i,t-j} - \sum_{j=o}^{\infty} Z_{ij}\ T_{o,t-j} \qquad (4)$$

where $Y_{i,j}$ and $Z_{i,j}$ are the response factors determined from the construction data. For the lightweight structures such as doors and windows

$$\frac{QK_{i,t}}{A_i} = \frac{QK_{o,t}}{A_i} = UT\ (T_{i,t} - T_{o,t}) = U\ (T_{a,t} - DB_t) \qquad (5)$$

where UT is the overall thermal conductance between the interior and exterior surfaces and U is the overall heat transfer coefficient between the indoor and outdoor air.

In addition to the above M equations (equation (1)) for each of M interior surfaces, the following overall heat balance equation for the room has to be solved for determining the room air temperature $T_{a,t}$

$$\sum_{i=1}^{M} HC_i \,(T_{i,t} - T_{a,t}) + RS_{a,t} + RL_{a,t} + RE_{a,t} + \rho C G_{L,t}\,(DB_t - T_{a,t}) + \rho C G_v\,(T_{v,t} - T_{a,t}) = 0 \tag{6}$$

where $RS_{a,t}$ = rate of solar heat coming through windows and convected into room air at time t

$RL_{a,t}$ = rate of heat from light convected into room air at time t

$RE_{a,t}$ = rate of heat from appliances and occupants convected into room air at time t

ρ = air density

C = air specific heat

$G_{L,t}$ = mass flow rate of outdoor air infiltrating into the room at time t

$G_{v,t}$ = mass flow rate of ventilation air at time t

DB_t = outdoor air temperature at time t

$T_{v,t}$ = ventilation air temperature at time t

With the assumption that the moisture condensation and absorption over the room surfaces and structures is small the room humidity is calculated by a simple energy balance:

$$REL_t + \rho G_{L,t} \lambda (W_{o,t} - W_{a,t}) + \rho G_v \lambda (W_{v,t} - W_{a,t}) = 0 \qquad (7)$$

where REL_t = rate of latent heat released into the room air by appliances and occupants at time t

λ = latent heat of evaporation of water

$W_{o,t}$ = humidity ratio of outdoor air at time t

$W_{v,t}$ = humidity ratio of ventilation air at time t

$W_{a,t}$ = humidity ratio of indoor air at time t

For the heat dissipated from occupants, which are included in Equations (1) and (6), a linear approximation may be used such that

$$\begin{aligned} &\text{Sensible heat release of time t in Btu/hr} = 10\,(100 - T_{a,t}) \\ &\text{Latent heat release of time t in Btu/hr} = 10\,(T_{a,t} - 60) \end{aligned} \qquad (8)$$

where $T_{a,t}$ is in °F. These equations are valid for a resting person (metabolic heat production = 400 Btu/hr) subjected to an air temperature between 60 °F and 100 °F. Below 60 °F all the heat loss can be considered as sensible and above 100 °F all the loss can be considered as latent. Psychrometric subroutines permit the calculation of indoor wet-bulb temperature or relative humidity with the room air humidity ratio $W_{a,t}$ solved for from Equation (7).

For Equations (1) and (6) the radiative and convective portions of the heat emitted from lights and equipment are assumed to be

Equipment, Appliances, and People	$RE_{a,t}$ = 50% of total
	$\sum_{i=1}^{M} RE_{i,t} \cdot A_i$ = 50% of total
Light	$RL_{a,t}$ = 50% of total
	$\sum_{i=1}^{M} RL_{i,t} * A_i$ = 50% of total

The radiation portion of this heat gain and 100% of the net solar heat gain (adjusted for the shading coefficient coming through the window) are assumed to be uniformly distributed and absorbed by the room interior surfaces. At present there is no provision to include the effect of shadow over the exterior surfaces of the building and window for adjusting the solar heat gain.

Although it is well known that air infiltration, $G_{L,t}$ is a function of outdoor wind velocity and air leakage opening characteristics of the room exterior, accurate relations for $G_{L,t}$ as a function of wind velocity and building structure are not available. Consequently it is assumed, for the time being, that the air infiltration can be simulated by simply supplying as input data a specified number of air changes per hour. The computer program presently has the capability of changing this on an hourly basis in order to simulate for example increasing the air infiltration at night for purposes of more effective cooling.

Although this particular computer simulation program for the indoor temperature and humidity has never been verified by experimental observation, many similar programs of the same basic concept have been known to predict the indoor temperature to within 2 °F*/.

2.1 Indoor Air Velocity

In order to evaluate indoor conditions for human comfort or physiological stress, it is necessary that indoor air velocity be estimated. The indoor air velocity is, however, believed to vary depending upon the air leakage rate, ventilation rate, thermal gradient of the room, basic design of the room, and any local air moving device such as portable fan.

According to Givoni, the indoor air motion could vary between 5 and 60% of outdoor wind velocity depending upon the nature of the air passage resistance through the room (2). Since the information available for predicting indoor air velocity as function of temperature distribution and wind velocity is very scanty, it is assumed in the present calculation that the indoor air velocity is less than 20 ft per minute. This assumption should yield an upper-bound value of comfort index or physiological index for summer conditions and is therefore safer to use than to overestimate the cooling due to elevated air velocities.

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*/ Concurrently a separate investigation is taking place at NBS to provide verification of the computer programs on several HUD Operation Breakthrough buildings.

In this part, a sample format for the compilation and presentation of pertinent building data relevant to the thermal performance of buildings is shown. Data are taken from the proposals of the 22 participating housing system producers (HSP) in the HUD Operation Breakthrough project. Since the 22 HSP's involved in that project represent the major housing system producers in the U.S. building industry, it is felt that data contained in their proposals will provide adequate and meaningful bases for the practice and the state of the art in housing design.

Up to the present time, data for one-third of the 22 HSP's have been compiled. Because of the different and numerous ways of presenting their design by the contractors, a computer program has been written to arrange, digest, and present the data in a unified manner for easy reference. Pertinent parameters associated with the structural components such as the U-value, weight per unit area, thermal mass, and the thermal time constant are also computed. They will be employed for the purpose of classification of the structural components and presented together with other data such as glass area, crack length, floor area, etc. The physical meaning and usefulness of these parameters will be discussed later. Additional parameters will be computed and listed if and when they appear to be of significance in providing information in the classification process. A separate report will be prepared giving complete detail of this process.

2.2 Mean Radiant Temperature

Since the room temperature calculation routine is providing temperatures for all the interior surfaces, it is an easy matter to determine the mean radiant temperature (MRT) of the space. Presently, however, the mean radiant temperature routine has not been included in the program and it is assumed that MRT is the same as the air temperature.

3. Classification of Building Data

One of the most important things that govern the indoor environment of a building is the "thermal performance" of the entire structure. Because of the periodically varying solar radiation and the large fluctuation in external environment temperatures, steady-state heat transfer through the enclosing walls into the interior space of a building rarely occurs, and the heat storage effects of the walls and the internal objects must be taken into account in order to estimate accurately the cooling load or the variations of the indoor environment. The overall heat transfer coefficient (U-value) of a wall, which gives a direct indication of the resistance to heat flow under steady-state conditions, is no longer the only major indicator of the heat flow into the inside environment under transient conditions. The mass of the wall, which governs the energy storage capacity, is also a pertinent parameter to be considered.

3.1 Type of Building Data Compiled

From the drawings and specifications submitted by each of the housing system producers in the HUD Operation Breakthrough project, the following data are collected for each of the different types and sizes of dwelling units contained in the drawings:

1. Method of construction of the exterior wall, party wall, interior partition, roof, ceiling/floor, windows, and doors
2. Floor area for each of the rooms in a dwelling unit
3. Type of walls and openings, and their areas for each of the rooms in a unit
4. Crack length of the windows and doors
5. Interior lighting design
6. Natural and mechanical ventilation data
7. Shading devices for the window areas

3.2 Method of Building Data Presentation

Data collected from the drawings and specifications are processed by the NBS UNIVAC 1108 computer. A sample printout for a typical dwelling unit is shown on the enclosed Tables 1 to 3. It should be pointed out here that the format of the printouts is tentative, and may be revised to a better form later if deemed necessary.

Table 1 gives a layer by layer description and the thermophysical properties of the materials used in the construction of the structural components. The order of the layers are from the outside of a room toward the inside for the side walls and from top to bottom for the roofs and floor/ceilings. The thermal property values are taken from the HSP's drawings and specifications from the 1967 ASHRAE Guide and Data Book. Very often data for the specific heat (C_p) of certain materials are not available, and a value of .200 Btu/lbm °F is then assumed. Since the value of C_p does not vary much for most of the building materials (of the order of .150 to .400 with a major portion of the materials in the lower range of the spectrum), the value of .200 is considered a good assumption.

Table 2 gives a room by room tabulation of the areas of the floor, ceiling, and the enclosing walls and openings of a room in a dwelling unit. The four sides of the room are labelled as S1, S2, S3, and S4. The orientation of the unit is not specified, since this is one of the factors that is influenced by site location, terrain, units group plan, etc. as much as by thermal performance consideration. For a group of attached units (single family attached, multi-family low rise or high rise), the unit at the ends or corner of the group is chosen since it has the largest amount of wall area exposed to the outside environment and consequently will probably result in the most severe indoor temperature variation or the largest cooling load among the group. Table 2 also gives the crack lengths of the windows and doors which are required in air infiltration specification. Data for lighting (as internal heat

source) and furniture (as internal heat sink) are not available at the present time, and will be incorporated into the table when available. In general, data contained in Table 2 are required by the computer program developed by the Environmental Engineering Section of the NBS Building Research Division for indoor temperature or load calculation purposes. It should be pointed out that to simplify the data read-off procedure, certain approximations such as the elimination of corners and the neglecting of closet space have been made. The data in Table 2 are, therefore, not an exact description of the actual floor plan in the HSP's drawings. However, these approximations will most probably not affect the evaluation of the thermal performance of the dwelling unit in any significant way.

Table 3 gives the values of various relevant parameters associated with and affecting the thermal performance of a building. They are computed from the data contained in Tables 1 and 2, and will be used in the classification of building structural components. Briefly, these parameters are:

1. Floor Area: The floor area represents the total amount of living space in a dwelling unit, and gives an indication of the size of the unit among the same type of dwellings.
2. Ratio of Exterior Wall Area to Floor Area: This ratio gives an indication on the complexity of the shape of the unit. For example, a simple rectangular unit will have a lower ratio than a more complex shaped unit.

3. Ratio of Window Area to Exterior Wall Area: This ratio indicates the relative amount of glass area in a unit which has a direct bearing on the amount of solar radiation transmitted directly into the interior space.

4. U-value: The U-value is the overall heat transfer coefficient including the surface air film resistance (at 7.5 mph wind outside and still air inside) under steady-state conditions. This is the major factor used by most designers in estimating the heating and cooling load of a building. As mentioned previously, under unsteady-state conditions the U-value is no longer the only controlling factor in heat flow through walls since it does not take into account the energy storage effect of the walls. However, it does give an indication on the amount of resistance the wall offers to daily average heat flow and therefore acts as one of the parameters in determining the overall heat transfer through the walls.

5. Thermal Mass, $\rho C_p L$: The thermal mass is a counterpart of the U-value and gives an indication on the energy storage capacity of the walls. It is computed from the following equation:

$$\rho C_p L = \sum_{i=1}^{N} \rho_i C_{pi} L_i \tag{9}$$

where ρi, C_{pi}, L_i are the density, specific heat, and thickness, respectively, of the i^{th} layer in a multilayer structure. N is the total number of layers in the structure.

6. Weight per unit surface area, ρL: Because of the small variations in the magnitude of the specific heat for most building materials, the heat storage capacity of a wall is almost directly proportional to the weight per unit surface area of the wall. This parameter is therefore closely related to the thermal mass of the wall. It has the advantage over the thermal mass in that a knowledge of the values of the specific heat which are not available for certain materials is not required. The weight per unit area of a wall is computed by

$$\rho L = \sum_{i=1}^{N} \rho_i L_i \tag{10}$$

7. Thermal Time Constant (TH.T.C.): The TH.T.C. is a function of the thermal diffusivities of the layers in a multilayer wall, and is defined as the heat stored per unit of heat transmitted. It is therefore a combination of the thermal mass and the thermal resistance of the wall. It is derived on the basis of the analogy between the heat flow represented by a thermal circuit, and the time constant of a R-C electric circuit, and is given by (2)

$$\text{TH.T.C.} = \sum_{i} (Q/U)_i \tag{11}$$

where

$$\left(\frac{Q}{U}\right)_i = (R_{os} + L_1/k_2 + \ldots + L_i/2k_i) \cdot (L_i \rho_i C_{pi}) \tag{12}$$

where R_{os} is the resistance of the surface air film and k_i is the thermal conductivity of the material of the i-th layer. In the case of an air space, L_i/k_i is substituted by the resistance of the air space. It should be pointed out that the TH.T.C. given by the above equation also takes into account the order the layers of materials are arranged. A change in the order of the arrangement will change the rate of heat flow through the wall under transient conditions. This is because that the outside surface temperatures of the walls with different orders of arrangement in their layers will be different during the transient states. For example, in the day time, the surface temperature of a wall with an insulation layer next to the external surface layer will be higher than a wall constructed of the same layers of materials but with the insulation layer placed further away from the external surface layer, resulting in more heat being transferred back to the external environment by convection and radiation, and consequently less heat being transferred into the interior space. It is noted that this phenomena is not accounted for by either the U-value or the thermal mass where the order of arrangement of the layers is immaterial.

3.3 Planned Work in the Future for Building Data Processing

After the completion of the survey of building data on the 22 housing system producers, a variety of selected units of different design will be used as input to the computer program, and the resulting indoor temperature variations (not air conditioned) or cooling load (if air conditioned) under the same external environment conditions will be compared. The influence of the various parameters on the indoor environment will be examined, and a classification of high (or large), medium, low (or small), or various ranges of numerical values, will be placed on those parameters that show a definite trend in relation to the variations in the indoor environment. Additional parameters such as shading factor, ventilation rate expressed in number of air changes per hour, and color of the exterior surfaces will also be examined. A set of tables based on the design of the 22 HSP's but applicable to similar types of design by other housing producers may then be constructed. The number of the various parameters used in the classification will be kept as small as possible while still adequately describing the structures in relation to the thermal performance of a typical dwelling unit.

4. Comfort and Physiological Indices

There are approximately fifteen prominent indices or groups of experimental data that have been presented to indicate man's comfort or discomfort when subjected to a certain environment. The majority are experimental in nature (i.e., human subjects have been exposed to a specified and controlled environment and asked to describe their feeling or at least have physiological measurements taken on them during the exposure). However, during the last several years there has been a great deal of interest in describing mathematically the thermal interaction between man and his environment, thus enabling one to predict what occurs or how one might feel in given surroundings. It is generally accepted that some form of skin temperature is a good measure of discomfort in cold environments whereas sweat rate, skin wettedness or some other factor related to sweat production is the best indicator of discomfort in warm environments. The material presented here will concentrate on application to warm environments. The remainder of this chapter will be devoted to a discussion of the most widely accepted of these indices.

4.1 Old Effective Temperature

Undoubtedly the most familiar of all comfort indices is the American Society of Heating, Refrigerating and Air-Conditioning Engineers' Effective Temperature. The scale was first established in the 1920's by Houghten and Yaglou (6). Subjects wearing clothing having an unspecified insulation value*/ were passed between test chambers in which the temperature, air velocity, and humidity were maintained at specified levels and were asked to describe their feeling immediately upon entering the new environment. The results are shown in Figure 2 where lines of constant thermal sensation are shown plotted on a standard psychrometric chart. The experiments were conducted for a large range of air velocities; however, the data shown here is for an air velocity of less than 20 fpm**/. The lines of constant thermal sensation were arbitrarily given numerical values corresponding to the dry-bulb temperature at which the line intersected the saturation curve (100% relative humidity). If the dry-bulb temperature were a sole indicator of thermal comfort according to this scale, the lines would run vertically. The data therefore indicates that

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*/ At the time this particular index was developed, the analysis of human comfort had not progressed to the point that insulation values for clothing ensembles were being given.

**/ For purposes of this study, the data for all the indices described will be for very nominal air velocities (i.e., 20 fpm or 0.1 m/sec) since higher velocities very rarely occur in residences.

as the relative humidity is increased, the temperature has to be decreased to maintain the same feeling. This is natural since cooling by sweat evaporation is restricted somewhat as the relative humidity increases.

This scale originally did not take into account the effect of radiation to or from room walls where their temperature is maintained at values quite different from the room air. However, a correction was proposed later (7) in order to account for this effect. The ET scale continues to be accepted as valid in the very hot region where man's cooling depends almost entirely on his sweat evaporation but has been abandoned as inaccurate in the comfort region or the region of most interest to this study.

As a result of the apparent disagreement between field tests and the lines of constant thermal sensation or effective temperature at the moderate temperatures, ASHRAE undertook an additional study at their research laboratory in 1960 and the results are shown in Figure 3. In this study (8), subjects wore light indoor clothing and were seated at rest. Room surfaces were held at the same temperature as the room air in all tests, and the air motion in the space was 20 fpm or less. The thermal sensation votes used in analyzing the test data were those which were cast after approximately three hour occupancy. These lines show the conditions under which subjects cast votes of 3, 4, 5, or 6, indicating sensations of slightly cool, comfortable, slightly warm, and warm respectively. It is evident from Figures 2 and 3 that the old effective temperature index predicts a considerably greater effect of relative humidity on the thermal

sensation feeling than do the lines determined by Koch, Jennings, and Humphreys (8). In defense of the old effective temperature, it may be explained that the two groups of data apply to different conditions. The effective temperature lines indicate a person's feeling immediately after entering the conditioned space, while the lines of Figure 3 are for people who have been in the conditioned space for approximately three hours and are in equilibrium with the environmental conditions.

4.2 Resultant Temperature

In 1948, Missenard (9) proposed an index called the resultant temperature. Its development was motivated by the fact that the effective temperature was not based on experiments where thermal equilibrium was achieved between the human body and the environment. The experiments on which the resultant temperature was based were conducted with about six subjects in a psychrometric chamber with a duration of exposure greater than was used in the experiments for the effective temperature. This index is identical in structure with the effective temperature and is shown in Figure 4 for clothed subjects exposed to an air velocity of 0.1 m/sec. The resultant temperature predicts a much less effect of thermal sensation on humidity than did the effective temperature. However, the results still indicate more of a dependence on humidity in the moderate temperature range than do the comfort votes of Koch, et.al.

4.3 Kansas State Index

Perhaps the most definitive experimental study on thermal comfort is one conducted on over 1600 college students at Kansas State University over the past several years (10, 11). The students were exposed (over a 3 hour period) to a uniformly heated or cooled environment (wall temperatures identical to the air temperature) with moderate air velocities and asked to vote on a scale from 1 to 7 which ranged from cold to hot:

1. cold
2. cool
3. slightly cool
4. comfortable
5. slightly warm
6. warm
7. hot

All subjects, males and females alike, were clothed in cotton twill shirts and trousers. The shirts were worn outside of the trousers. The subjects wore the underwear they had on when reporting for this test, which usually consisted of cotton undershorts or jockey shorts for the men and brassieres and underpants for the women. All other clothing, including T-shirts, girdles, slips, etc., were removed. The subjects wore cotton sweat socks without shoes. The net insulating value for the clothing ensemble, as computed by means of an electrically heated copper manikin (12) has been determined to be 0.6 clo[*/] which has been generally accepted as the standard clothing for thermal comfort studies.

[*/] 1 clo represents a measured resistance to heat transfer of the clothing ensemble of 0.18 sq. m hr °C/Kcal (0.88 sq. ft hr °F/Btu or equivalent of an overall heat transfer coefficient = 1.13 Btu per (hr) (sq. ft) (F))

Even though the results are available for men and women separately and also for the votes taken at each half an hour for the entire 3 hours of exposure, the data shown in Figure 5 is an average for the votes of men and women taken together at the end of the 3 hours. Rohles (13) discusses the fact that a single line on a temperature grid to represent a "comfortable" thermal sensation provides little in the way of practical information unless some type of limits surround the line or some border is presented which differentiates the "slightly cool" conditions from the "comfortable" conditions and in turn the "comfortable" dimension from the limits of the "slightly warm" conditions. Consequently, the data was reanalyzed and the modal comfort envelopes shown in Figures 6 and 7 were presented. As long as the temperature and humidity gave a condition within the 70% envelope, at least 70% of the subjects voted 4 (comfortable). The same interpretation goes with the 60% envelope. It should be pointed out that over 90% of the subjects voted either 3, 4, or 5 (slightly cool, comfortable, or slightly warm) when the space condition was anywhere within the 70% envelope.

4.4 Predicted Mean Vote and Predicted Percentage of Dissatisfied

One of the most significant attempts to predict analytically the thermal interaction between man and his environment is presented by P. O. Fanger of the Technical University of Denmark in his recent book Thermal Comfort (14). He begins by pointing out the most important variables which influence the condition of thermal comfort:

1. activity level (heat production in the body)
2. thermal resistance of the clothing (clo-value)
3. air temperature (dry-bulb)
4. mean radiant temperature*/ (to account for the radiation transfer between the subject and the surroundings)
5. relative air velocity
6. water vapor pressure in ambient air

The basis of his analysis in the so-called general comfort equation which defines all combinations of the variables which will create thermal comfort. Its specific form is as follows:

$$M - E_{diff} - E_{sw} - E_{res} - D = K = R + C \qquad (13)$$

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*/ Mean radiant temperature is defined as the uniform surface temperature of an imaginary black enclosure with which man (also assumed a black body) exchanges the same heat by radiation as in the actual environment.

where M = the internal heat production in the human body

E_{diff} = the energy loss by water vapor diffusion through the skin

E_{sw} = the energy loss by evaporation of sweat from the surface of the skin

E_{res} = the latent respiration heat loss

D = the dry respiration loss

K = the heat transfer from the skin to the outer surface of the clothed body (conduction through the clothing)

R = the heat loss by radiation from the outer surface of the clothed body

C = the heat loss by convection from the outer surface of the clothed body

The double equation expresses that the internal heat production M minus the heat loss by evaporation from the skin ($E_{diff} + E_{sw}$) and by respiration ($E_{res} + D$) is equal to the heat conducted through the clothing K and dissipated at the outer surface of the clothing by radiation and convection (R + C). It is assumed that the evaporation corresponding to E_{sw} and E_{diff} takes place at (or underneath) the skin surface.

After formulating expressions for each component in the above equation (see Appendix A) and substituting them into Equation 13, two algebraic equations result in the following 9 unknowns:

M - (defined previously)

T_s - skin surface temperature

P_w - water vapor pressure in the ambient air

E_{sw} - (defined previously)

T_a - ambient air dry bulb temperature

T_{cl} - clothing surface temperature

I_{cl} - insulation value of the clothing

T_{MRT} - mean radiant temperature of the environment

v - the relative air velocity

Six of the unknowns (M, P_w, T_a, I_{cl}, v, and T_{MRT}) can be considered as controllable or specified for any particular subject--environment situation. Two of the remaining unknowns are eliminated by what may be considered Fanger's most questionable assumption. During studies at Kansas State University on college students, data was obtained for mean skin temperature and evaporative heat loss of the subjects while they reported being comfortable conducting various activities (different M's). E_{sw} and T_s were plotted individually as a function of the metabolic production and even though the data was rather widely scattered, linear equations

$$T_s = \text{fct}\ (M) \tag{14}$$

and

$$E_{sw} = \text{fct}\ (M) \tag{15}$$

were determined by regression analysis. The procedure then used was to specify 5 of the 6 controllable or specified quantities (M, P_w, T_a, I_{cl}, v, and T_{MRT}), and use the 4 equations (13, 14 and 15) to determine the remaining four unknowns, one of which is the sixth of the controllable parameters that would give thermal comfort. The results are presented in the form of comfort charts.

Having determined the conditions for optimum thermal comfort, Fanger recognized the need for being able to evaluate an existing room climate and quantify its deviation from the comfort condition. A similar scale to the one used in the Kansas State studies was chosen:

-3	cold
-2	cool
-1	slightly cool
0	neutral
+1	slightly warm
+2	warm
+3	hot

The numerical values however are lower by 4. A scale is thus obtained which is easier to remember, as it is symmetrical around the zero point, so that a positive value corresponds to the warm side and a negative value to the cold side of neutral. He then proposed that the thermal sensation values, to be called the Predicted Mean Vote (PMV), would be a function of the thermal load of the body, defined as the difference between the internal heat production and the heat loss to the actual environment for a man hypothetically kept at the comfort values of the mean skin temperature and the sweat secretion of the actual activity level.

$$PMV = fct\ (M - E_{diff} - E_{sw} - E_{res} - D - R - C) \qquad (16)$$

The nature of Equation 16 enabled Fanger to use the same component expressions as in the general comfort equation. The particular form of the function was determined from analyzing the Kansas State studies once again as well as similar experiments conducted at the Technical University of Denmark. The exact expression corresponding to Equation 16 is given in Appendix A.

Values of the predicted mean vote (PMV) are shown on the psychrometric chart in Figure 8. The data was calculated assuming a resting metabolic rate ($M = 50$ kcal/hr m^2), an air velocity of 0.1 m/sec, a mean radiant temperature equal to the ambient air temperature, and a clothing insulation value of 0.6 clo. In Figure 9 the results are compared with the experimentally determined KSU comfort vote. As can be seen the agreement is excellent for the comfort line (KSU = 0, PMV = 4) but not so good for environmental conditions outside of the comfort range.

An advantage of an analytical model such as this is the capability of observing the effect of one of the six controllable parameters without having to conduct extensive experiments. For example, it may be unrealistic to determine an air conditioning criteria using subjects that wear the standard 0.6 clo clothing ensemble. Certainly in hot weather people would be more inclined to wear lighter clothing such as shorts and an open neck shirt with short sleeves. Data for this type of clothing ensemble (clo = 0.25) is shown in Figure 10 and compared with the standard Kansas State clothing ensemble in Figure 11. The comparison shows what one might intuitively expect. In the ligher clothing, subjects would report comfort at 4 or 5 °F warmer temperature. It would

be expected that they would vote cool at a somewhat higher temperature also (approximately 6 °F). However, both subjects would vote hot at approximately the same temperature and relative humidity.

Instead of just giving the predicted mean vote as an expression for the thermal environment, Fanger felt it might be even more meaningful to state what percent of persons can be expected to be decidedly dissatisfied since this can readily be interpreted by both the engineer and the layman. This same concept has been proposed most recently by Nevins and McNall (3). After choosing the limits of -2 and +2 for being dissatisfied on the cold and hot side respectively, the American and Danish experiments already mentioned were again analyzed to determine the percent of the subjects voting cool or below and warm or above at each environment condition. The same relationship was assumed to hold between PMV and the new factor, predicted percentage of dissatisfied (PPD). The exact relation is given in Appendix A and some results using the PPD are shown in the next chapter.

4.5 New Effective Temperature

The other major analytical contribution to the field of thermal comfort to be mentioned in this paper is a transient heat transfer model presented by Gagge, Stolwijk, and Nishi (15). The model differs most significantly from Fanger's in the fact that physiological responses have been included such as blood flow restriction and dilation to and from the skin and regulatory sweating proportional to a deep body temperature deviation from some predetermined set point. The human body is pictured as composed of two distinct parts; a central core and a

skin shell. An energy balance is written for each part:

$$\begin{bmatrix}\text{the rate of}\\ \text{energy in-}\\ \text{crease of}\\ \text{the skin shell}\end{bmatrix} = \begin{bmatrix}\text{the net rate}\\ \text{of energy}\\ \text{transfer from}\\ \text{the core to}\\ \text{the skin}\end{bmatrix} + \begin{bmatrix}\text{the net rate}\\ \text{of energy}\\ \text{transfer from}\\ \text{the environ-}\\ \text{ment to the}\\ \text{skin}\end{bmatrix}$$

and

$$\begin{bmatrix}\text{the rate}\\ \text{of energy}\\ \text{increase}\\ \text{of the}\\ \text{central}\\ \text{core}\end{bmatrix} = \begin{bmatrix}\text{the rate}\\ \text{of energy}\\ \text{generation}\\ \text{(metabolic}\\ \text{production)}\\ \text{in the core}\end{bmatrix} + \begin{bmatrix}\text{the net rate}\\ \text{of energy}\\ \text{transfer from}\\ \text{the skin to}\\ \text{the core}\end{bmatrix} + \begin{bmatrix}\text{the net rate of}\\ \text{energy transfer}\\ \text{from the environ-}\\ \text{ment to the core}\end{bmatrix}$$

or in equation form

$$M_{sk} C_{sk} \frac{dT_{sk}}{d\tau} = Q_{con} + Q_{bl} - E_{diff} - E_{rsw} - R - C \tag{16}$$

and

$$M_{cr} C_{cr} \frac{dT_{cr}}{d\tau} = M - Q_{con} - Q_{bl} - E_{res} \tag{17}$$

where

M_{sk} = mass of the skin shell

C_{sk} = specific heat of the skin shell

T_{sk} = temperature of the skin shell

M_{cr} = mass of the central core

C_{cr} = specific heat of the central core

T_{cr} = temperature of the central core

τ = time

Q_{con} = heat transferred by conduction from the core to the skin (a function of T_{cr} and T_{sk})

Q_{bl} = energy transferred by blood flow from the core to the skin (a function of T_{cr} and T_{sk})

E_{diff} = the energy loss by water vapor diffusion through the skin

E_{rsw} = the energy loss by evaporation as a result of regulatory sweat production (a function of T_{cr} and T_{sk})*/

R = the heat loss by radiation from the skin to the environment

C = the heat loss by convection from the skin to the environment

M = metabolic heat production in the core

E_{res} = the latent respiration loss from the lungs

The form of the differential equation implies the lumped capacity analysis or in other words, the temperature of the skin and core are assumed uniform throughout at every instant. The model is no doubt unrealistic in this respect, since the temperature can and does often vary widely within the body.

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*/ This particular component is extremely important in the performance of the model yet the expression doesn't agree with all physiological observations reported in the literature. A discussion of this discrepancy is given in Appendix B.

After formulating expressions for the various energy components and choosing appropriate numerical values for the various parameters (see Appendix A), the two differential equations are solved simultaneously (numerically on a digital computer) to predict the time variation of the core and skin temperatures when the controllable parameters mentioned previously are specified. The model predicts a response that levels off at the end of approximately one hour. Once T_{cr} and T_{sk} have been determined, it is possible to work back through the analysis and calculate the regulatory sweat production (at the end of one hour), associated with the given environmental-human conditions. Mass transfer theory is then used to calculate the maximum possible evaporation cooling for given skin temperature and ambient temperature and vapor pressure. The ratio of calculated regulatory sweat evaporation to the maximum possible is given the notation of skin wettedness. It is then posulated (as a result of experimental observation) that in warm environments, various conditions giving the same skin wettedness will also cause the same thermal sensation or feeling of warmth or coolness. This particular point is debatable and is the subject of much discussion in Appendix B.

Lines of constant skin wettedness at the end of one hour's exposure are shown in Figure 12 for a resting subject (58.2 watts/m^2) with the standard clothing (0.6 clo) subjected to a small air velocity. This model has been proposed to ASHRAE and accepted as properly predicting the lines of constant thermal sensation. Consequently the lines in Figure 12 are labelled as lines of New Effective Temperature. Actually the model predicts zero regulatory sweating at 77 °F and 50% R.H. or in other words, the 77 ET lines corresponds to a skin wettedness of 0. Effective temperature lines below this temperature are simply drawn as parallel to the 77 ET line. The numerical values associated with the lines were determined by using the dry bulb temperature at the intersection of the constant skin wettedness line and the 50% relative humidity curve. This in contrast to the original effective temperature where experimentally determine lines of constant thermal sensation were given numerical values corresponding to the dry bulb temperature where the lines intersected the saturation or 100% relative humidity curve. The labelling of the new effective temperature seems more reasonable since the values will more closely correspond to dry bulb temperatures experienced in everyday living. However, the confusion that may result after publication of the 1972 volume of the ASHRAE Handbook of Fundamentals can be seen by observing Figure 13. The numerical values of the two scales are very widely separated. Since the old effective temperature was considered standard for almost 50 years, many guidelines and comfort limits will have to be reevaluated in terms of the new scale.

By observing the new scale, one sees that the dependence of thermal sensation on relative humidity at the higher temperatures is approximately the same as with the old effective temperature. This is quite acceptable since the old effective temperature scale was considered accurate at these temperatures. However at temperatures near the comfort zone, the new model predicts a much less dependence on relative humidity which is in good agreement with the most recent studies. These two factors no doubt contributed significantly to the new scale's acceptance by ASHRAE. Published with the new scale in the 1972 Handbook will be a comfort zone and a broad comfort zone shown in Figures 14 and 15 respectively.

4.6 Heat Stress Index

The heat stress index was developed by Belding and Hatch at the University of Pittsburgh and first published in 1955 (16). The postulation that forms the basis of their index is as follows: the metabolic production of the body minus the energy that can be given off by radiation and convection to the surroundings equals a quantity of energy that must be given off by sweat evaporation in order to maintain thermal equilibrium with the environment. Consequently the expression for the required evaporative cooling (E_{req}) is

$$E_{req} = M - R - C \qquad (18)$$

As mentioned previously, for a given skin temperature and ambient air temperature and vapor pressure, mass transfer theory can be used to calculate the maximum evaporative cooling possible. The heat stress

index is defined as being equal to the ratio of the two quantities E_{req}/E_{max} multiplied by 100. Belding and Hatch assumed a constant skin temperature of 35 °C and used the most up to date (at that time) expressions for radiation and convection heat loss from the body (see Appendix A) to calculate values of the index and present them in nomographs as functions of the metabolic production and environmental conditions. In addition, physiological and hygienic implications of 8 hour exposures to various heat stresses were given and are shown in Table 4.

Calculations have been made using the original expressions for radiation and convection losses and maximum evaporative cooling from the body for a resting person subjected to a uniform environment with an air velocity of 20 fpm. The results are shown in Figure 16 in the form of lines of constant heat stress index. Notice that using these expressions sweat evaporation to maintain body equilibrium is not required until the environment reaches approximately 82 °F. This is in definite contrast to Gagge's effective temperature model which shows regulatory sweat production starting at approximately 5 °F lower. The expressions were reevaluated and updated in 1966 by McKarns and Brief (17). ASHRAE plans to recommend in their 1972 Handbook of Fundamentals the calculation of the index using the same expressions for R, C and E_{max} that were used in the new effective temperature model. In addition, expressions for dry and latent heat losses from the lung passages have been subtracted from the right side of Equation 18 and the results of new calculations are shown in Figure 17 and compared with the original lines of constant heat stress in Figure 18. The latest computations

seem to be reasonable since the evaporative cooling as a result of sweat production is seen to be necessary at a temperature as low as 74 °F. Figure 19 shows that a heat stress index of 10 which corresponds to mild heat strain agrees quite well with the upper side of the modal comfort envelopes as proposed by Rohles (13).

As the new computations for heat stress index were being carried out, special note was made of the addition of expression for dry and latent heat losses from the lung passages to Equation 18. This seems quite reasonable since this cooling occurs naturally and would certainly assist in maintaining body thermal equilibrium. However, since latent cooling due to water vapor diffusion through the skin also occurs naturally the computations were carried out once again subtracting this factor (E_{diff}) from the right side of Equation 18. The results are shown in Figure 20 and it can be seen that they predict evaporative cooling due to regulatory sweating necessary at approximately 77 °F (50% relative humidity) which agrees almost identically with the new effective temperature model. The contribution that the three "natural" cooling processes make can be seen in Figure 21 where on the one hand only the new expression for R and C were used in Equation 18 and on the other the expressions for the three "natural" cooling processes were also used in the equation.

In retrospect, it can be seen that the quantity called skin wettedness in the new effective temperature model is very similar to a heat stress index. Instead of dividing the maximum possible evaporative cooling rate into a required evaporative cooling rate, the maximum value is divided into a sweat cooling rate that is predicted from known physiological responses. In line with this thinking, the results of Gagge's model are replotted in Figure 22 and interpreted as another heat stress index.

4.7 Thermal Comfort When Equilibrium is Maintained by Sweating

Before publishing the new effective temperature model, Gagge and his associates published still another model for indicating thermal comfort (18) that is very similar to the heat stress index. The cooling due to evaporation of sweat production necessary to maintain body thermal equilibrium is determined by subtracting the body heat loss by radiation, convection, latent loss from the lung passages and the loss due to evaporation of moisture diffusing through the skin from the metabolic heat production. (See Appendix A for the specific relations.) The resulting required evaporative cooling is divided by the maximum possible and then multiplied by 100 to give a factor "percentage of wettedness".

The main difference between this model and the two previous ones mentioned is that the skin temperature is not held constant as in the case of heat stress index or predicted from known physiological responses as with the new effective temperature model, but rather assumed to be a polynomial function of the environment temperature (see Appendix A). The function was determined by fitting an equation (by regression analysis) to experimental data that was determined at the John B. Pierce Foundation. Computations were made using the model and the results are shown in Figure 23 for a resting person with the standard clothing (0.6 clo) subjected to a uniform environment with an air velocity of 20 fpm. Since the model is so similar in structure to the heat stress index and the skin wettedness factor of the new effective temperature model, all three are compared in Figure 24. Since all three use the same expressions for R, C, and E_{max}, the discrepancies can be attributed to the three different ways in which the skin temperature is determined and also to the absence or inclusion of the "natural" cooling terms (see Appendix A).

4.8 Index of Thermal Stress

Givoni (2, 19) has done extensive work over the past few years on measuring and evaluating heat stress. Having compared actual response with the predictions of many models and found much disagreement, he developed his own "index of thermal stress" (ITS) which according to additional extensive experiments conducted by Givoni correctly predicts the physiological strain imposed on resting and working people by metabolic and environmental factors. The index is structured exactly like the heat stress index with two important differences. An expression has been included for the radiant heat load due to solar radiation so that the index can predict responses for people working or resting outdoors. This is not directly applicable to this study but certainly makes a useful addition to the field of heat stress evaluation. The other difference is a factor called cooling efficiency of sweating (f) and is included in the equation as follows:

$$E_{req} = [M - R - C]\, \frac{1}{f} \qquad (19)$$

The concept is that where other indices assume all the sweat produced or required is evaporated and provides cooling for the body, this may not be true. For example, some of the evaporation could occur in the clothing and not result directly in cooling for the body. Calculations using Givoni's expressions (see Appendix A) have been made and are shown in Figure 25. The data used was for a resting person ($M = \frac{100\ \text{kcal}}{\text{hr}}$), subjected to a uniform environment with an air velocity of .1 m/sec. The clothing ensemble chosen (affects the radiation and convection

expression) was very similar to the Kansas State standard clothing. It should be noted that the upper limit for comfort (according to Givoni) is associated with an evaporative loss of 60 grams/hr and that distinct thermal discomfort is experienced progressively as the sweat rate is elevated above 100 grams/hr.

Even though Givoni states that the model predicts a sweat rate that is in agreement with experimentally measured rates, the results shown in Figure 25 appear questionable. Sweating is seen necessary at approximately 64 °F. The index does show small dependence on relative humidity at the lower temperatures and somewhat more dependence at the higher temperature which agrees with the new effective temperature model. However, it should be remembered that what is shown in the figure is sweat rate and not the ratio between the sweat rate and some maximum possible. To put it in a form comparable to the heat stress index, values of the above mentioned ratio were calculated and are shown in Figure 26 and compared with the original ITS values in Figure 27.

4.9 Predicted Four Hour Sweat Rate

The P4SR index was developed by McArdle and colleagues during World War II at the Royal Naval Research Establishment (20). Sweat rate was measured for fit, young men dressed in shorts only or overall and shorts when subjected to many different combinations of temperature, humidity, and air velocities for four hour periods. The results were presented in the form of nomographs and have been adapted to a psychrometric chart as shown in Figure 28. The particular conditions for this chart are a

metabolic production of 54 kcal/m^2 hr, a uniform radiation environment (mean radiant temperature equal to air temperature), and an air velocity of 10 fpm. As was the case with the original heat stress index, sweat production is shown to start only after the temperature has reached approximately 80 °F. Based on the experiments, a limit of 3.0 was established as the upper tolerable limit for daily exposure.

4.10 Wet-Dry Index

In 1957 Lind and Hellon (21) proposed the use of the Oxford Index or wet-dry index to assess the severity of hot climates. It is simply a combination of the wet-bulb and dry-bulb temperatures (°F) according to

$$WD = 0.85\ T_{wb} + 0.15\ T_{db} \qquad (20)$$

The extreme dependence on relative humidity is obvious and is shown in Figure 29. Due to its simplicity it has been used quite often in setting limits or correlating data in extremely hot and wet conditions where there is no radiant stress and very low air velocity.

4.11 Temperature-Humidity Index

Another index very similar to the Oxford Index is one called the temperature-humidity index defined by

$$THI = 0.4\ (T_{ab} + T_{wb}) + 4.8 \qquad (21)$$

where all values are in °C. This index like the last, attempts to combine the effect of temperature and humidity in environments where radiant stress and air velocity are not important. Figure 30 shows that the THI index is much less dependent on humidity than the W-D index was. As an example of its use, Bridger and Helfand (22) correlated death rates during the 1966 heat wave in St. Louis and found that when the 24 hour average of the THI was greater than 27 °C (81 °F), there was an usually large number of deaths due to heat stroke.

The authors have intended this part to be simply a review and presentation of important heat stress and comfort indices that could be integrated into the air conditioning criteria study. There is no clear cut choice where one index has been universally accepted as correctly predicting physiological responses in all situations. On the contrary, there is considerable disagreement on the relative merit of the various indices. However a general method of attack can be suggested. Indices showing very slight dependence on humidity in the probable comfort range of 75 to 80 °F (i.e., KSU, PMV, new ET) could be used for setting limits on comfort conditions. In addition, the ones showing strong dependence on humidity at the higher temperatures (i.e., new ET, W-D, HSI) could be used for setting upper bounds for prevention of serious health hazards.

Other important factors should be noted. Most of the indices are valid for, or at least have been developed for, quasi-steady situations. The subjects have been exposed to (or the models simulate the subjects being exposed to) steady state environments for periods of time up to four hours and their response is given accordingly. Very seldom does this occur in the average home or office building and the technique of extending these indices to a more transient environment is questionable. ASHRAE, in their comfort standard, recommend limiting the rate of change of dry bulb temperature in a space to 4 °F/hr and the rate of change of relative humidity to 20%/hour. These limits have been verified somewhat by an experimental study at Kansas State University reported by Sprague and McNall (23). College students wearing the standard clothing were subjected to an environment where the dry bulb temperature and relative humidity were oscillated one at a time about the comfort value while the other parameter was maintained at the comfort value. The conclusion of the study was that the comfort standard values are somewhat conservative but no change was recommended.

Another factor to be considered quite important is the effect of nonuniform environments. There is a capability built into many of the indices to handle a situation where the wall temperatures (all considered the same) are different than the ambient air temperature. However, one parameter such as the mean radiant temperature cannot possibly describe actual environments where perhaps every surface is different in temperature and may not even be uniform on the individual surfaces. In addition, very little or nothing is known about the physiological response to this kind of environment. All of the considerations mentioned above simply point to the tremendous amount of work to be done in this field.

5. Feasibility Study for the Determination of an Air Conditioning Criteria

In order to determine whether the approach mentioned in Part 1 is feasible for determining a meaningful air conditioning criteria, an analysis depicted in Figure 31 has been carried out. An apartment to be built in Macon, Georgia and Jersey City, New Jersey as part of HUD's "Operation Breakthrough" (Camci) has been simulated in the National Bureau of Standards' computer program (NBSLD) described in Part 2. Required as input was all pertinent building data for the apartment as well as hour by hour values of important weather parameters. Hourly values of outdoor dry-bulb temperature, wet-bulb temperature, wind velocity, and cloud cover were taken from the United States Weather Bureau tapes (available from the National Climate Center in Ashville, North Carolina) for the months of June, July, August and September, 1949 through 1958. The output of the program was hourly values of

indoor dry-bulb and wet-bulb temperature for the four month, ten year period. Algorithms for pertinent physiological indices were written along with detailed statistical subroutines so that daily profiles and certain histograms showing values of these indices could be obtained.

The building data that was used is as follows:

1. The apartment building was assumed to face west and the individual apartment chosen was one at approximately mid-height of the 130 feet high structure. No building shadow was used and the adjacent apartments and hallway were assumed to always be at the same temperature as the apartment.
2. The exposed wall is to be constructed of 7 inches of concrete and 2 inches of insulation. It contains a 100 ft^2 glass window (shading coefficient = 0.22 corresponding to a double pane window with a white opaque roller shade). The total wall area is 212 ft^2 leaving 112 ft^2 of solid structure.
3. The floor/ceiling unit is composed of 5 1/2 inches of concrete and 1/8 inch cork tile with a total area of 550 ft^2. The floor to ceiling height is 8.5 ft.
4. The partitioned walls contain 6 inches of concrete.
5. The maximum number of occupants is 2, the maximum equipment load is 1/2 watts/ft^2 of floor area, and the maximum lighting load is 3 watts/ft^2 of floor area.

6. The infiltration or natural ventilation rate was simulated as 1 air change/hr between 6 a.m. and 7 p.m. and 6 air changes/hour from 7 p.m. to 6 a.m.

Typical results for the four months of one summer in Jersey City are shown in Figures 32 through 67. The data is for the summer of 1954 which was perhaps the coolest of the 10 year period. The wet-bulb temperature was above 67 °F for only 583 hours and the dry-bulb temperature was above 80 °F for only 493 hours of the four month summer. Shown in Figure 32 is a plot of the solar energy falling on the apartment (after modification by actual cloud cover) and a plot of outdoor dry and wet-bulb temperature for the month of June. Figure 33 shows the output of the thermal simulation of this apartment; in other words, a monthly profile of indoor dry and wet bulb temperature. Since the ASHRAE comfort standard gives limits on the rate of change of indoor temperature and relative humidity, these values were computed and are shown in Figure 34. As can be seen, the suggested values of 4 °F/hr and 20%/hr were only exceeded once or twice during the month and this was also found true for every other summer month of the ten year period in Jersey City and Macon, regardless of how hot the weather was. Figures 35 through 40 show monthly profiles of several of the indices discussed in Part 4. Comments that can be made concerning these profiles are as follows:

1. The new effective temperature was higher than the upper limit of the comfort zone (81) 8.7 percent of the time and higher than the upper limit of the broad comfort zone (85) only 1 percent of the time during the month.
2. The predicted mean vote for a person in standard clothing (0.6 clo) exceeded 1 (slightly warm) 4.2 percent of the time but was above 1 only 1.1 percent of the time when the clothing was reduced (0.25 clo).
3. There was practically little or no heat stress regardless of the particular index used (old or new).
4. The Kansas State Index was above 5 (slightly warm) only 1.0 percent of the time.
5. More than 10 percent of the people were dissatisfied only four times during the month with light clothing but that increased to 7 or 8 times with standard clothing assumed.

The same kinds of observations could be made about the three remaining months from Figures 41 through 67.

In addition to the summer profiles of the various parameters, certain statistical data was gathered and examples are shown in Table 5 and Figures 68 through 70. The table shows for the 10 summers in Jersey City, the number of total hours in each summer that the outdoor dry bulb temperature exceeded 80 °F and the outdoor wet-bulb temperature exceeded 67 °F. In addition, the percent of the time in each month that selected indices exceeded prescribed levels was calculated. Since not only the total length of time but also duration of each hot period will affect the general satisfaction or dissatisfaction with the space conditions, the length of time that each index stayed above the prescribed level for each hot period was computed. In Figure 68 for example, the new effective temperature was above 85 for a total of 14 percent of the time during August of 1953. The specific way it occurred was: once for 2 hours, twice for between 12 and 18 hours and once for more than 48 hours. Contrast this with July of 1958 (Figure 69) where the total amount of time was approximately the same but the manner in which it happened was entirely different: 4 times for only 2 hours at a time, 5 times for 3 hours, once for 6 hours, twice for 7 hours, twice for 8 hours, once for 10 hours, and finally once for between 12 and 18 hours. Most probably, persons would not express the same degree of dissatisfaction during these two months.

It is felt that given the kind of information just shown for a specific structure and location the decision to air condition or not could certainly be made. However it would be neither practical nor economical to undertake such an analysis for each individual case. It may not even be feasible to carry out the analysis for all possible combinations of building groups and climatic zones across the United States. Therefore a short-cut method was sought by examining the results of this thermal simulation very closely.

In Figure 71 through 76 it can be seen that correlations were attempted between time (in 4 months) above a prescribed index value inside and time the outdoor dry-bulb or wet-bulb temperature exceeded 80 °F or 67 °F respectively. Even if successful, the result would naturally have only been valid for the specific apartment in the specific location. However it was felt that this would be one way to describe quantitatively the "thermal performance" of the unit. As the figures indicate, the number of hours in several consecutive months that the dry-bulb temperature is above 80 °F or the wet-bulb temperature is above 67 °F does not correlate well with what occurs inside the apartment during this same period of time. These parameters of course are precisely the ones used in the present HUD criteria.

Seeking a correlation covering a shorter period of time, Figures 77 and 78 show the daily maximum values of the KSU index inside as a function of maximum dry-bulb temperature and wet-bulb temperature outside for August of 1954 in Jersey City. The results are much better than the previous ones but still too scattered to be of much value. However, when the same dependent variable was plotted as a function of average (over a 24 hour period) outdoor dry-bulb temperature the result was excellent as shown in Figure 79. The correlation coefficient (r) for this particular set of data was 0.97185 (maximum possible value = 1.0). The same degree of success was attained with other indices such as the new effective temperature shown in Figure 80. The daily average temperature used was not the average of the twenty-four hourly values but rather the average of the daily maximum and daily minimum. This particular average was chosen due to the fact that many United States Weather Bureau Stations report precisely this parameter[*/]. The discrepancy is usually quite small as the curves of Figure 81 indicate.

[*/] Instructions for Climatological Observers Circular B, U. S. Weather Bureau Publication, January, 1962.

The straight lines of Figure 79 and 80 were determined by regression analysis (method of least squares) in order to fit the data as close as possible. In fact, similar regression lines were determined for June, July and August of all ten years in both Jersey City and Macon. It was found that each of the 30 lines for Jersey City fell within the corridor of Figure 82 and similarly in Macon (Figure 83). As would be expected the changing of any major building parameter would cause the apartment to "perform" differently. The effect of only reducing the shading over the large window can be seen in Figure 84 and similarly in Figure 85 for keeping everything the same except the facing direction of the exterior wall.

The importance of such correlation results lie in the fact that if proven valid by additional analytical and/or experimental studies a particular line could be determined for a given housing unit-weather data group by a small number of calculations compared to a 4 month 10 year hour by hour computation. The decision to air condition would then have to be made not on the total time or percent of time over an extended period that an index value was exceeded but rather whether the daily average outdoor temperature for some given design conditions caused an indoor environment that was unacceptable when compared with prescribed levels of the PIHI.

To compare an approach such as this with the ten year computation already presented, additional calculations were carried out. The 1% design values of outdoor dry-bulb and wet-bulb temperature presented in the ASHRAE Handbook of Fundamentals (24) for several selected cities (see Table 6) were used in conjunction with the dimensionless daily temperature cycle of Figure 86 (determined from surveying actual weather tapes) to produce a design day cycle for these cities. The Camci apartment was then subjected to this design day cycle (in the computer program) for seven straight days using the actual solar radiation in these localities and assuming zero cloud cover. Shown in Figure 87 are the results of the calculations where the maximum value of the KSU index in the apartment during the second and fifth day are plotted for each city. There was no noticeable change after the fifth day. As can be seen, the agreement between the data points and previously determined corridors is quite good.

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Appendix A

Component Expressions for Various Comfort Models and Ranges of Application

In Part 4, the methods of computation for several of the analytically determined comfort indices were discussed. Since most of them use expressions for heat transfer rate between the subject and environment by convection, radiation, etc., these expressions are given here for purposes of comparison.

<u>Rate of Heat Loss by Convection (C)</u>

PMV: $C = A_{DU}\ f_{cl}\ h_c\ (T_{cl} - T_a)$ kcal/hr

$$h_c = 2.05\ (T_{cl} - T_a)^{0.25} \text{ for } 2.05\ (T_{cl} - T_a)^{0.25} > 10.4\ \sqrt{v}$$

$$= 10.4\ \sqrt{v} \text{ for } 2.05\ (T_{cl} - T_a)^{0.25} < 10.4\ \sqrt{v}$$

ET (new): $C = h_c\ (T_{sk} - T_a)\ F_{cl}$ watts/m^2

$$h_c = 11.6\ v^{0.5} \text{ watts/m}^2\ {}^\circ\text{C}$$

HSI (original): $C = 2\ v^{0.5}\ (95. - T_a)$ Btu/hr

where v = air velocity, fpm

T_a = ambient air temperature, °F

HSI: identical with ET (new)

Thermal Comfort when Equilibrium is Maintained by Sweating: identical with ET (new)

ITS: $C = \alpha v^{0.3} (35 - T_a)$ kcal/hr

<u>Rate of Heat Loss by Radiation (R)</u>

PMV: $R = 4.8 \times 10^{-8} A_{DU} f_{cl} f_{eff} [(T_{cl} + 273)^4 - (T_{MRT} + 273)^4]$ kcal/hr

ET (new): $R = h_r (T_{sk} - T_a) F_{cl}$ watts/m^2

HSI (original): $R = 22 (95 - T_{MRT})$ Btu/hr

T_{MRT} = mean radiant temperature, °F

HSI: identical with ET (new)

Thermal Comfort when Equilibrium is Maintained by Sweating: identical with ET (new)

ITS: accounted for in the convection term by replacing T_a by T_{MRT}

<u>Rate of "Dry" Heat Loss in Respiratory Passages (E_{res})</u>

PMV: $E_{res} = .0014 M (34 - T_a)$ kcal/hr

ET (new): zero

HSI (original): zero

HSI: $E_{res} = .0014\ M\ (34 - T_a)$ watts/m^2

M = metabolic heat production, watts/m^2

Thermal Comfort when Equilibrium is Maintained by Sweating: zero

ITS: zero

<u>Rate of Energy Loss due to Evaporative Cooling in Respiratory Passages (E_{res})</u>

PMV: $E_{res} = .0023\ M\ (44 - P_a)$ kcal/hr

ET (new): $E_{res} = .0023\ M\ (44 - P_a)$ watts/m^2

M = metabolic heat production, watts/m^2

HSI (original): zero

HSI: identical with ET (new)

Thermal Comfort when Equilibrium is Maintained by Sweating: identical with ET (new)

ITS: zero

Rate of Energy Loss due to Evaporative Cooling, Normal Skin Diffusion (E_{diff})

PMV: $E_{diff} = .35\ A_{DU}\ (43 - 0.061\ \frac{M}{A_{DU}} - P_a)$ kcal/hr

ET (new): $E_{diff} = 0.06 \cdot 2.2\ h_c\ (P_{sk} - P_a)\ F_{pcl}$ watts/m^2

h_c = convection coefficient given above, watts/m^2 °C

HSI (original): zero

HSI: identical with ET (new)

Thermal Comfort when Equilibrium is Maintained by Sweating:

$E_{diff} = .35\ (P_{sk} - P_a)\ F_{pcl}$ watts/m^2

ITS: zero

Rate of Energy Loss due to Evaporative Cooling, Regulatory Sweat Production (E_{sw}, E_{rsw})

PMV: $E_{sw} = .42\ A_{DU}\ (\frac{M}{A_{DU}} - 50)$ kcal/hr

ET (new): $E_{rsw} = 70\ (T_{cr} - 36.6)\ (T_{sk} - 34.1)$ watts/m^2

for $T_{cr} \geq 36.6$ °C and $T_{sk} \geq 34.1$ °C

$E_{rsw} = 0.0$

for $T_{cr} < 36.6$ °C or $T_{sk} < 34.1$ °C

HSI (original): not applicable

HSI: not applicable

Thermal Comfort when Equilibrium is Maintained by Sweating: not applicable

ITS: not applicable

<u>Maximum Evaporative Cooling Rate Possible (E_{max})</u>

PMV: not specified

ET (new): $E_{max} = 2.2\ h_c\ (P_{sk} - P_a)\ F_{pcl}$ watts/m^2

h_c = convection coefficient given previously, watts/m^2 °C

HSI (original): $E_{max} = 10\ v^{.4}\ (P_{sk} - P_a)$ Btu/hr

v = air velocity, fpm

HSI: same as ET (new)

Thermal Comfort when Equilibrium is Maintained by Sweating: same as ET (new)

ITS: $E_{max} = P\ v^{.3}\ (42 - P_a)$ kcal/hr

Rate of Heat Conduction Through Clothing (K)

This particular quantity is only applicable to PMV and is given by:

$$K = A_{DU} \frac{T_{sk} - T_{cl}}{.18\ I_{cl}} \text{ kcal/hr}$$

Relation Between PMV and Body Heat Load (L)

$$PMV = [.352\ e^{-.042\ (M/A_{DU})} + .032]\ L$$

Relation for Sweating Efficiency

$$\frac{1}{f} = e^{0.6\ (E_{req}/E_{max} - 0.12)}$$

Relation for PPD

A second order equation was fit to the tabular data given by Fanger for the Predicted Percentage of Dissatisfied (due to heat only) as a function of PMV

$$PPD = 1.63 + 10.98\ (PMV) + 13.41\ (PMV)^2$$

Relation for Skin Temperature as a Function of Ambient Air Temperature

The following equation was used in Gagge's "Thermal Comfort When Equilibrium is Maintained by Sweating" model

$$T_{sk} = 25.49 + .249\ T_a \ °F$$

Unless it was otherwise specified, the following nomenclature applies to the relations:

A_{DU} = body surface area, m^2

α = clothing coefficient (approximately 11.6 for 0.6 clo standard clothing)

f_{cl} = ratio of the surface area of the clothed body to the nude body (= 1.1 for 0.6 clo standard clothing)

F_{cl} = a factor that measures the efficiency for the passage of dry heat from the skin surface through the clothing to the environment (= .58 for 0.6 clo clothing)

f_{eff} = the ratio of the effective radiation area of the clothed body to the surface area of the clothed body (0.65 for seated position and 0.75 for standing)

F_{pcl} = permeation efficiency factor for water vapor evaporated from the skin surface through clothing to the ambient air (= 0.82 for 0.6 clo clothing)

h_c = convection coefficient, kcal/hr m^2 °C

h_r = radiation coefficient, watts/m^2 °C, "= 5.23 in the ET model"

I_{cl} = conduction resistance value for clothing, clo

M = metabolic heat production, kcal/hr

P = clothing coefficient (approximately 13.0 for standard clothing)

P_a = vapor pressure in ambient air, mm Hg

P_{sk} = saturated vapor pressure corresponding to skin temperature, mm Hg

T_a = ambient air temperature, °C

T_{cl} = clothing surface temperature, °C

T_{cr} = core or deep body temperature, °C

T_{MRT} = mean radiant temperature, °C

T_{sk} = skin surface temperature, °C

v = air velocity, m/sec

In addition to the relations, Table 7 has been included to give the range of application of the various indices as given on psychrometric charts in this report.

Appendix B

Discussion of Regulatory Sweating Models

In Part 4, it was noted that the component expression for regulatory sweating in the new effective temperature model contributed significantly to the response of the model. As given in Appendix A, the component expression is

$$E_{rsw} = 70\ (T_{cr} - 36.6)\ (T_{sk} - 34.1)\ \text{watts/m}^2$$

$$\text{for } T_{cr} \geq 36.6\ °\text{C and}$$

$$T_{sk} \geq 34.1\ °\text{C}$$

$$E_{rsw} = 0 \text{ if } T_{cr} < 36.6\ °\text{C or}$$

$$T_{sk} < 34.1\ °\text{C}$$

The interpretation of this expression is that sweat production is caused by signals from both the skin and the deep body. If the temperature in either one of the regions rises above the normal value, then the body secrets sweat for the purpose of evaporative cooling. This sweating model is in definite constrast to the results of experiments conducted by Dr. T. H. Benzinger of the Naval Medical Research Institute (25, 26). Dr. Benzinger concludes as a result of his studies, that the signal for sweat secretion is a sole function of the temperature in the anterior portion of the hypothalamus region of the brain. When this temperature rises above a sharply determined set point, warm sensitive neurons begin to fire and excite sweating such that the rate of sweat production is

directly proportional to deviation beyond the set point.

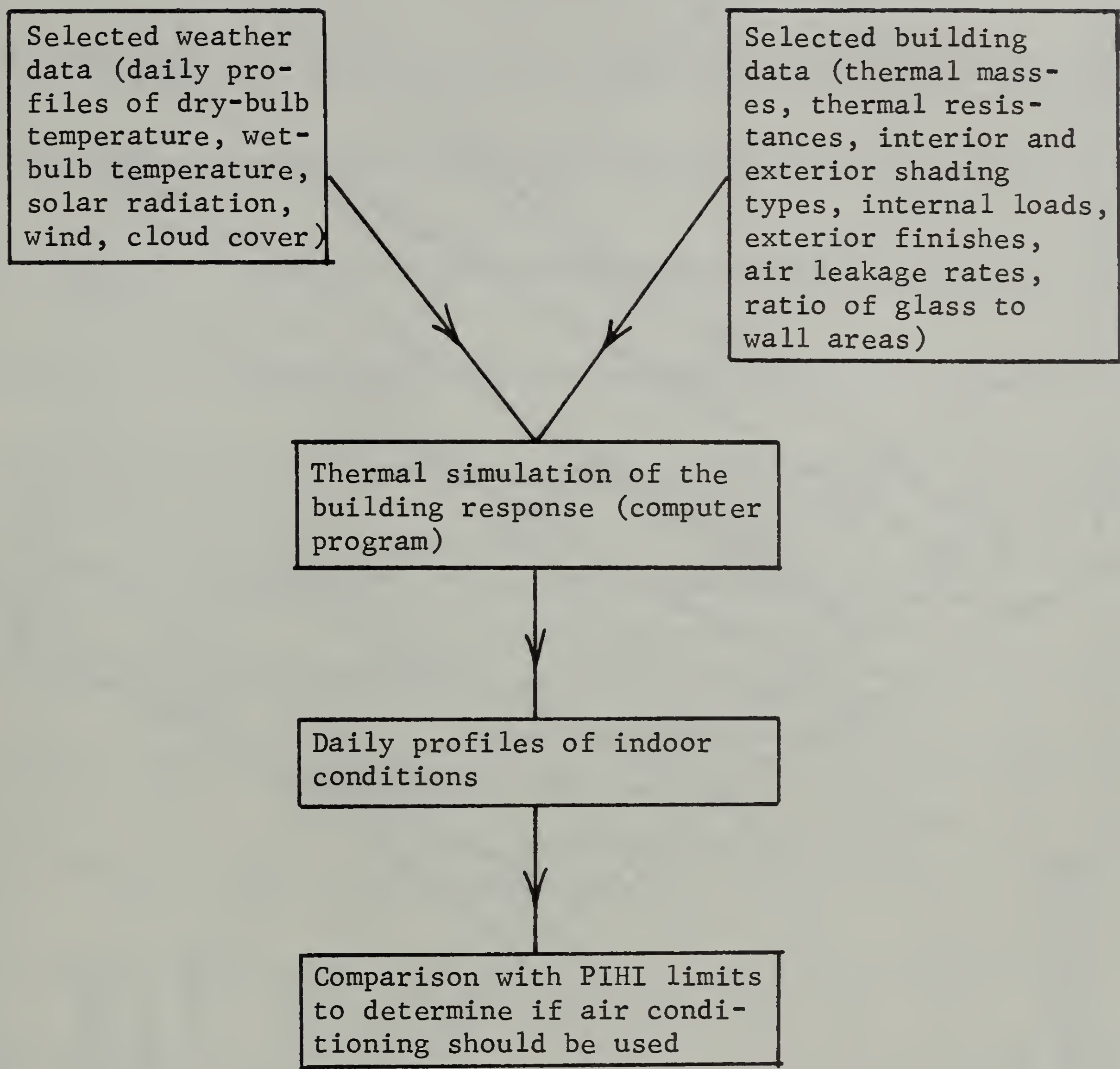
The skin temperature does play a role in the evaporative cooling process; however, in contrast to the interpretation of Hardy and Stolwijk (27, 28) (which has been adapted for the new effective temperature model), Benzinger concludes that the effect is one of inhibition. Whenever the skin temperature drops below approximately 33 °C the rate of sweat production begins to decrease below its normal value (given a hypothalamus temperature above the set point) and the amount of decrease is proportional to the deviation of skin temperature below 33 °C. A comparison between these two distinctly different interpretations is shown in Figure 88. The rate of evaporative cooling is plotted against the deviation of skin temperature from 33 °C. As can be seen, Gagge's model simulates zero cooling by sweat evaporation as long as the skin temperature is below 34.1 °C and is directly proportional to $(T_{sk} - 34.1) \cdot (T_{cr} - 36.6)$ above this point. Benzinger's data indicates a constant rate of sweat production whenever the skin temperature is above 33 °C (the value of the rate dependent on $(T_{cr} - 36.6)$) and a progressively decreasing rate as the skin temperature drops below 33 °C.

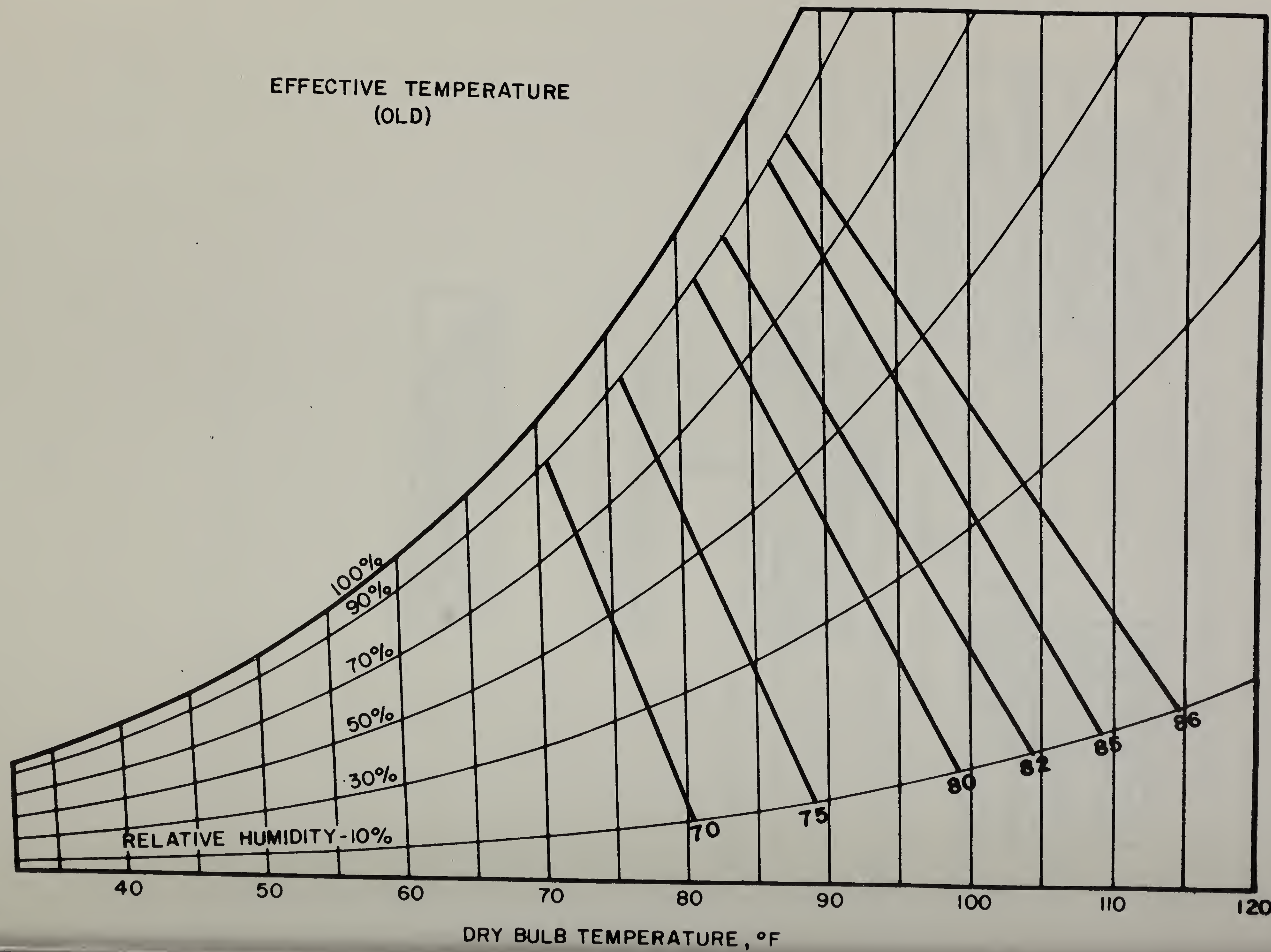
The importance of the evaporative cooling in the model can be seen in Figure 89 where new effective temperature lines are plotted that have been calculated using Benzinger's results for sweat production. Since his data generally gives larger values of evaporative cooling for specified core and skin temperatures, it would be expected that this modified scale would show more dependence on humidity than the standard new effective temperature lines and this can be seen in Figure 90. It is not recommended that the new data be used in the simulation of the body response since the experiments used to generate them were very special in nature. Men were enclosed in a gradient calorimeter (see reference 29) such that all sweat evaporated could be instantly carried away. This may not be true in ordinary environments, particularly where the humidity of the ambient air is high. However, it should be obvious that much more study and refinement is needed before an accurate simulation of human response to an environment can be accomplished.

The interpretation that lines of equivalent thermal comfort are identical with lines of constant sweat ratio could be adjusted in light of Dr. Benzinger's findings. The feeling of thermal comfort is no doubt linked to the necessity for sweat secretion when subjected to warm environments. However, as a result of his studies this is in turn linked directly to the hypothalamus temperature or deep body temperature since this is the sole source of the sweating signal. The reasoning is substantiated by an additional experiment conducted by him. A subject was cooled in a tank of cold water until the hypothalamus temperature was well below the individual's set point. He was then put into a very hot bath where the skin temperature and hypothalamus temperature were monitored

with time. Even though the skin temperature went to a relatively high value almost immediately, the subject did not report discomfort until the hypothalamus temperature began to rise above his set point.

The new effective temperature model as presently structured was used to generate lines of constant core temperature and these are shown in Figure 91. The numerical values assigned to the lines were again determined by the dry-bulb temperature along the 50% relative humidity curve. As would be expected, the lines don't show quite as much dependence on humidity as before and this can be seen in Figure 92. However, the agreement with the studies at Kansas State University are quite good as shown in Figure 93. In fact, the 85 ET line which was recommended as the upper limit of the broad comfort zone is identical to the KSU = 5 (slightly warm) line.


Selected weather data (daily profiles of dry-bulb temperature, wet-bulb temperature, solar radiation, wind, cloud cover)
Selected building data (thermal masses, thermal resistances, interior and exterior shading types, internal loads, exterior finishes, air leakage rates, ratio of glass to wall areas)
Thermal simulation of the building response (computer program)
Daily profiles of indoor conditions
Comparison with PIHI limits to determine if air conditioning should be used



EFFECTIVE TEMPERATURE
(OLD)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
82
85
86
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

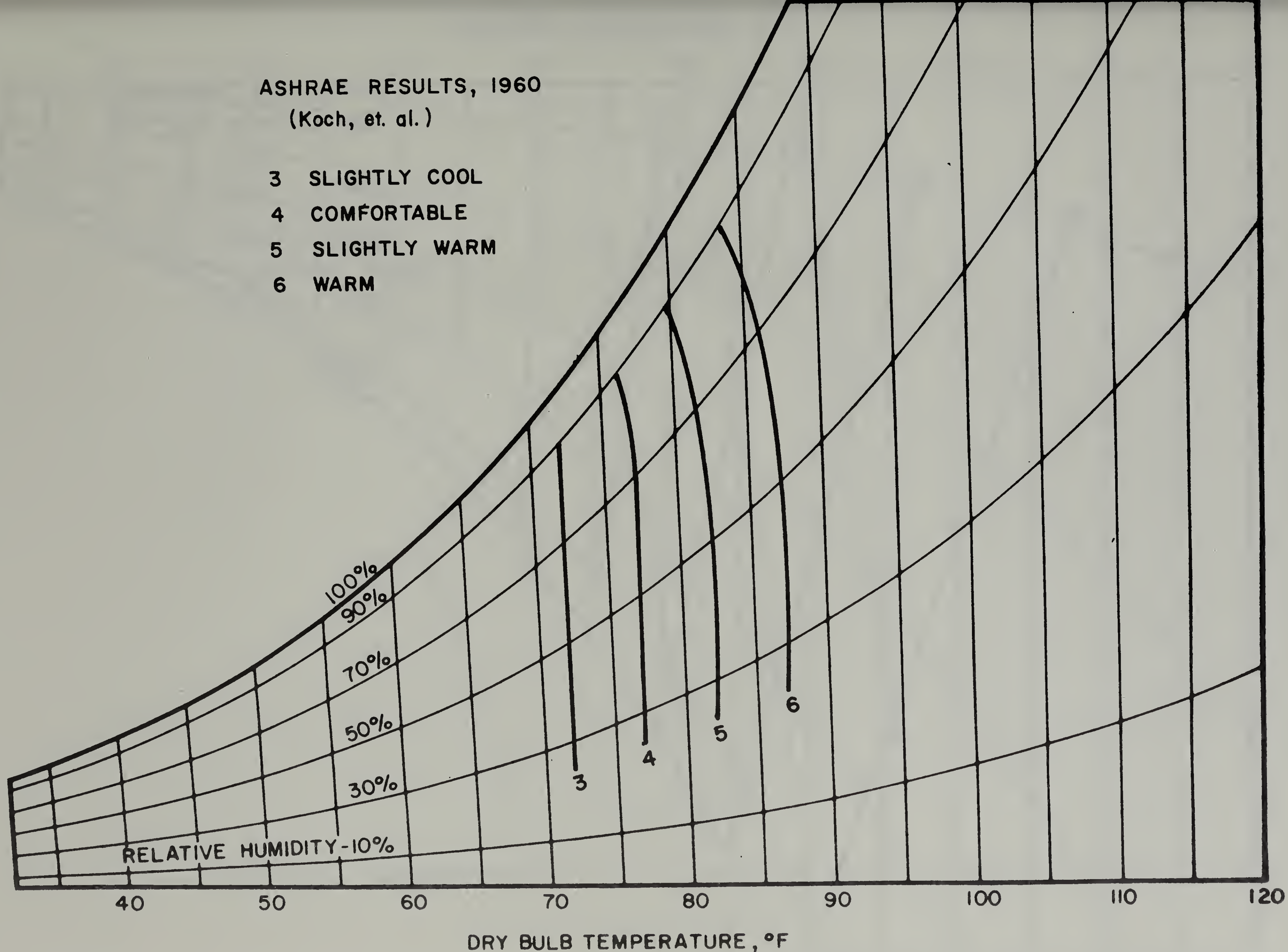

ASHRAE RESULTS, 1960
(Koch, et. al.)
3 SLIGHTLY COOL
4 COMFORTABLE
5 SLIGHTLY WARM
6 WARM
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
3
4
5
6
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 3

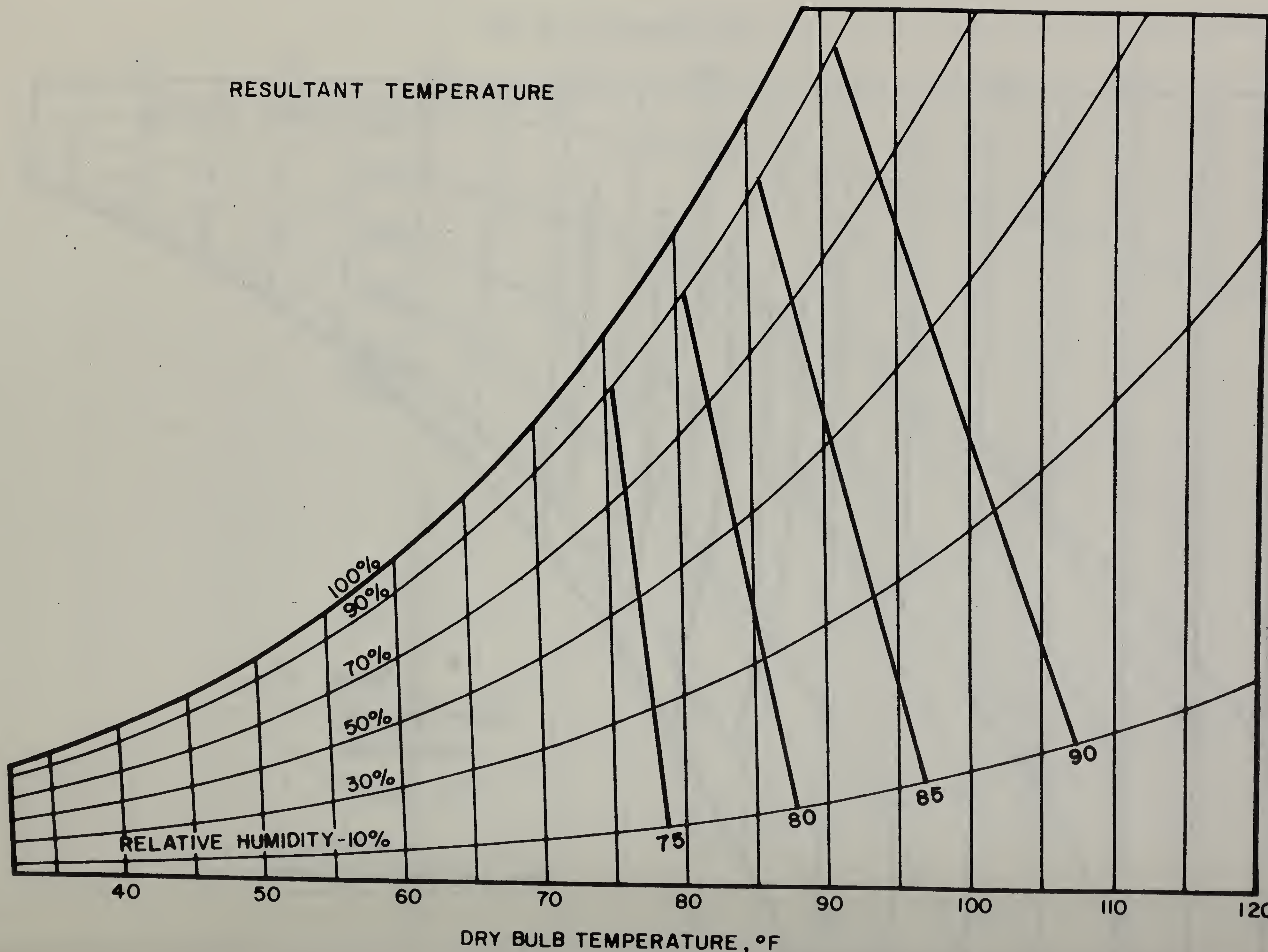

RESULTANT TEMPERATURE
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 4

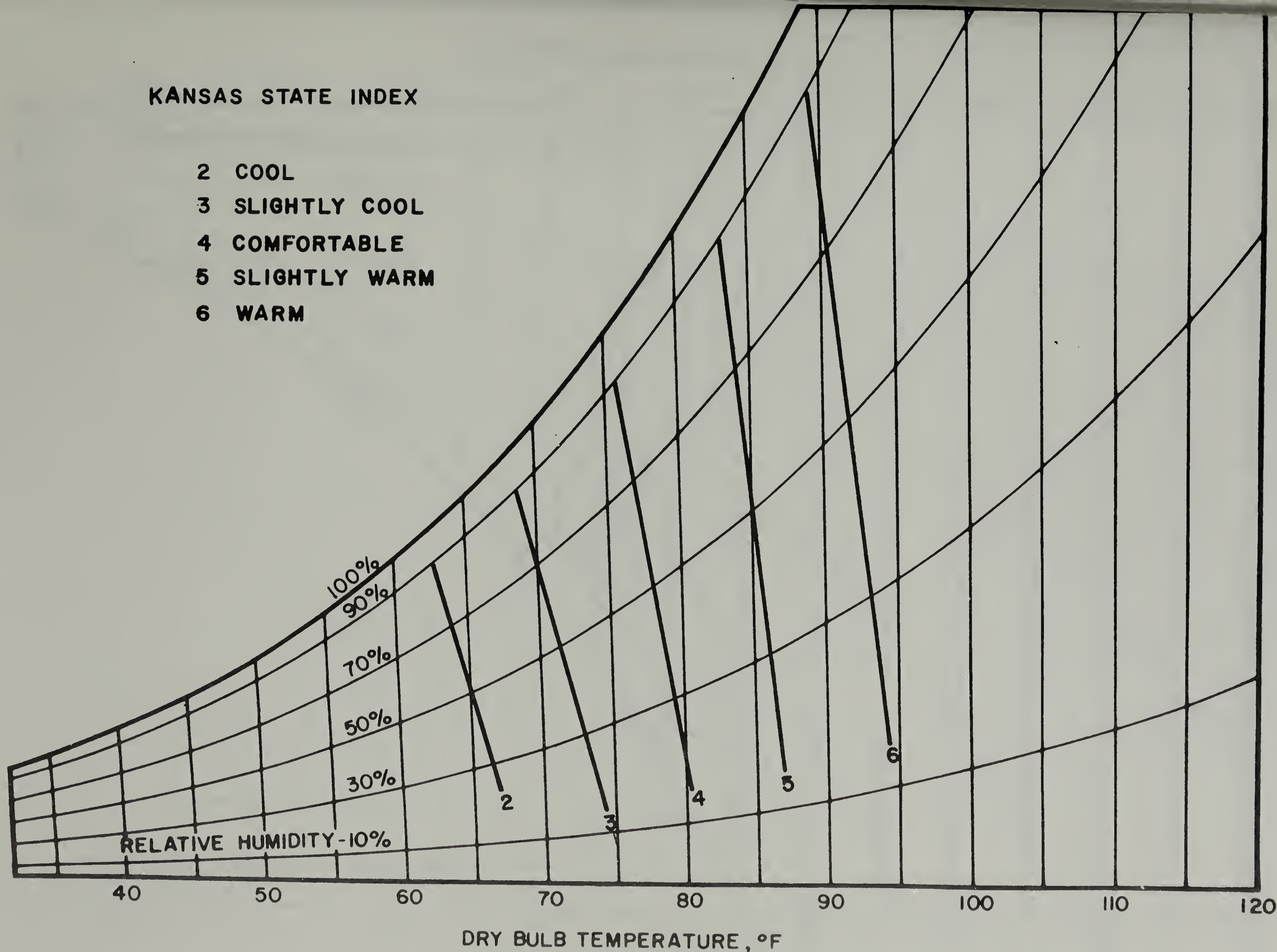

KANSAS STATE INDEX
2 COOL
3 SLIGHTLY COOL
4 COMFORTABLE
5 SLIGHTLY WARM
6 WARM
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
2
3
4
5
6
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 5

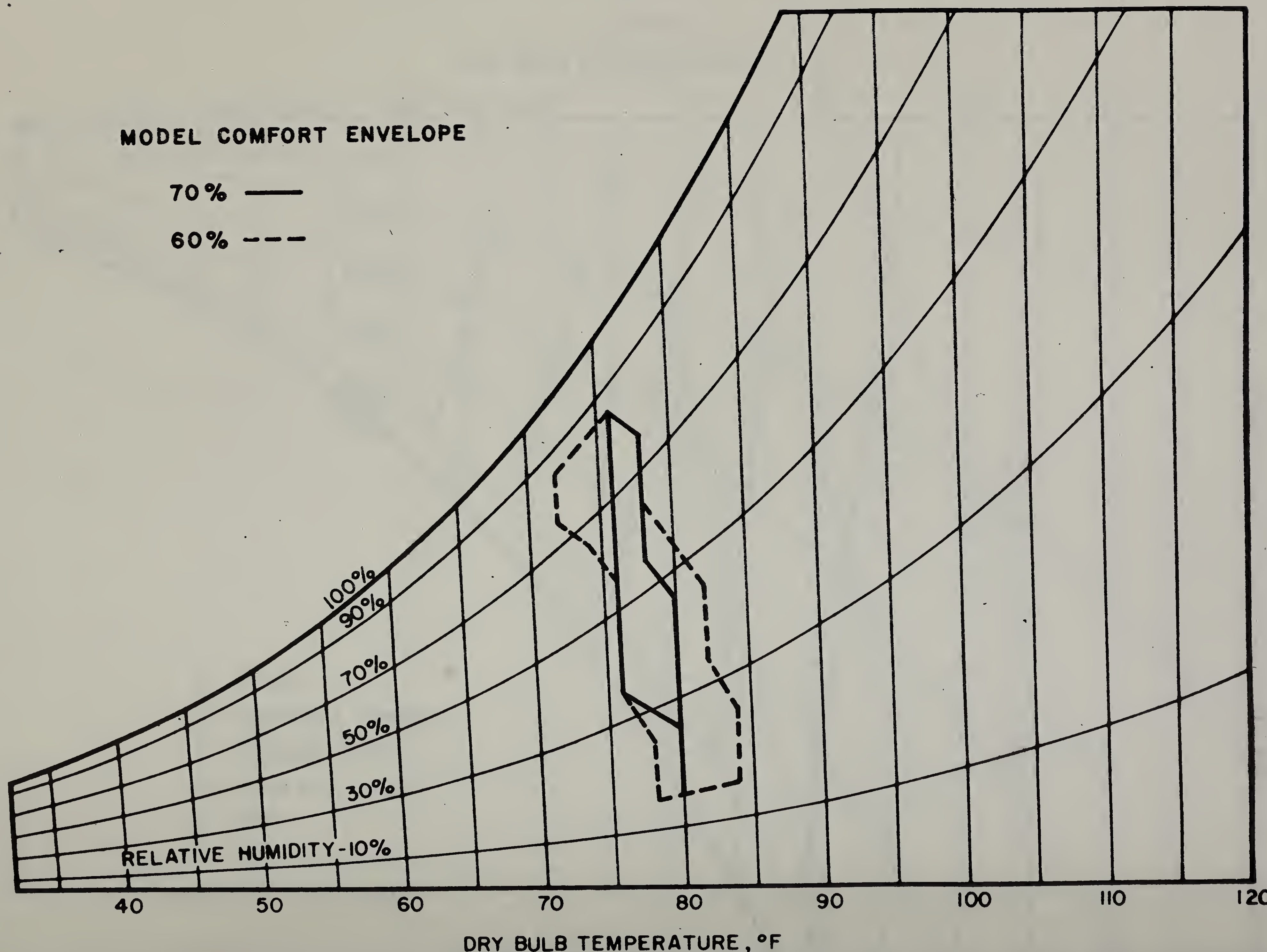

MODEL COMFORT ENVELOPE
70% ———
60% - - -
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 6

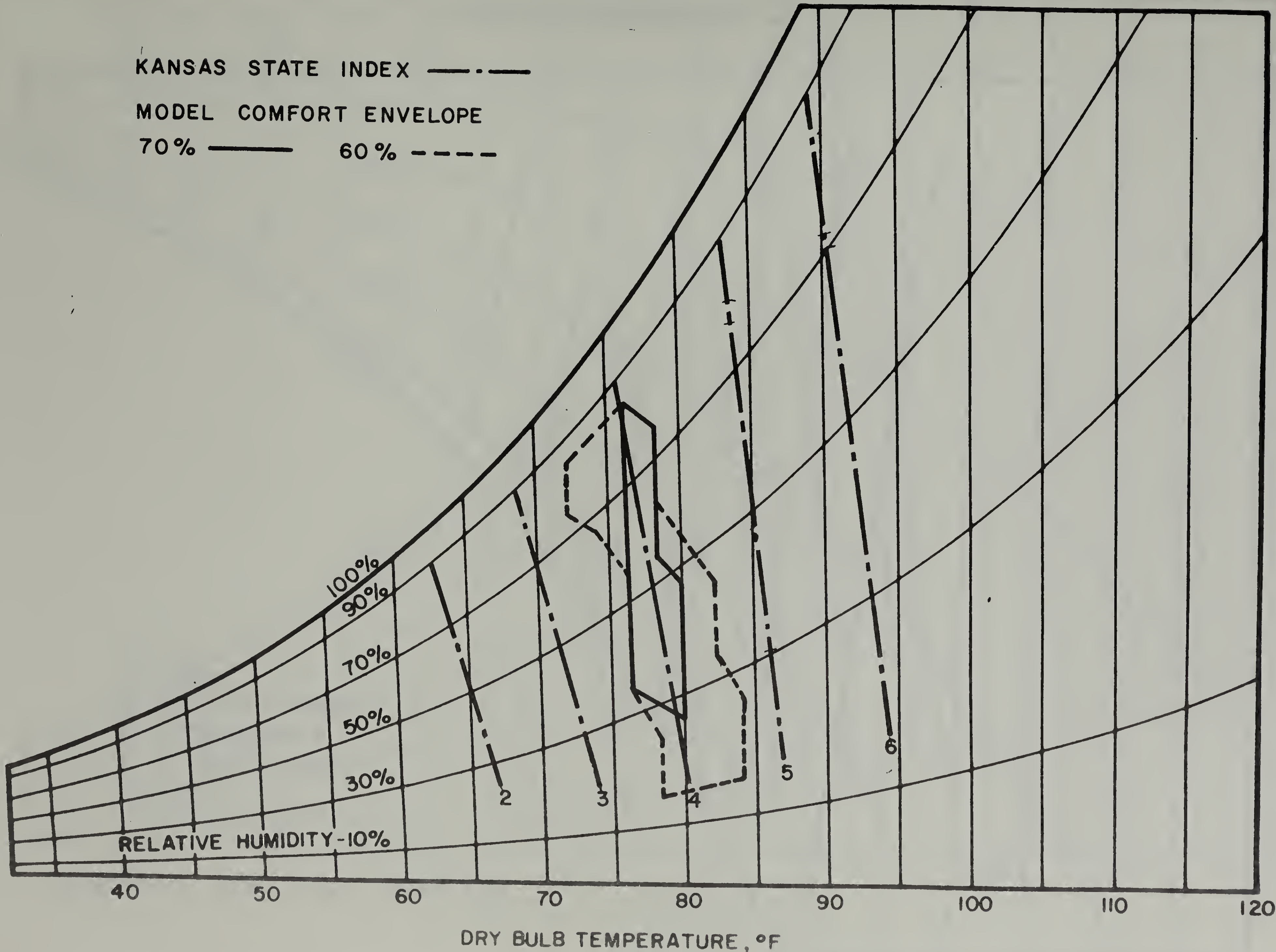
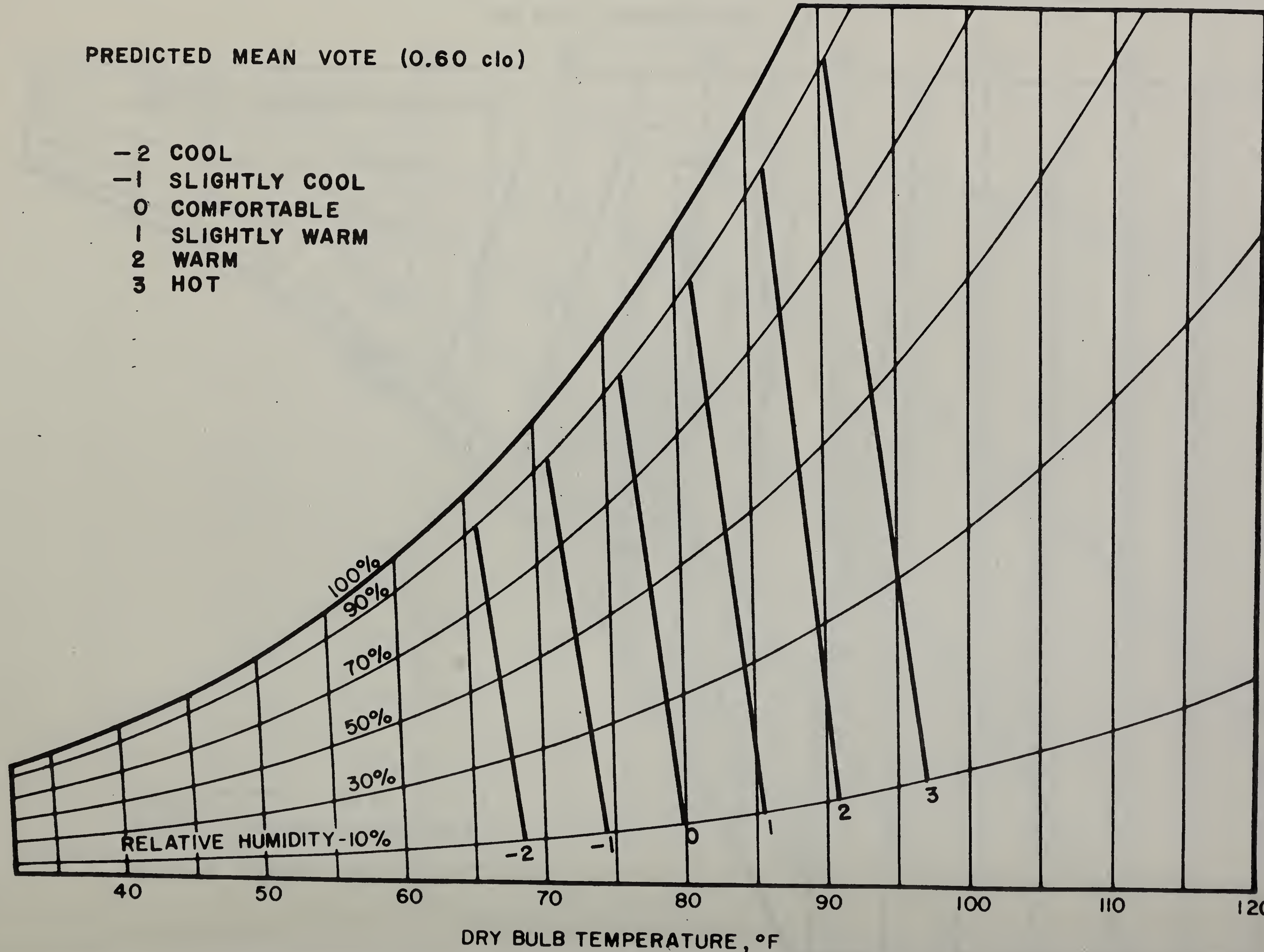
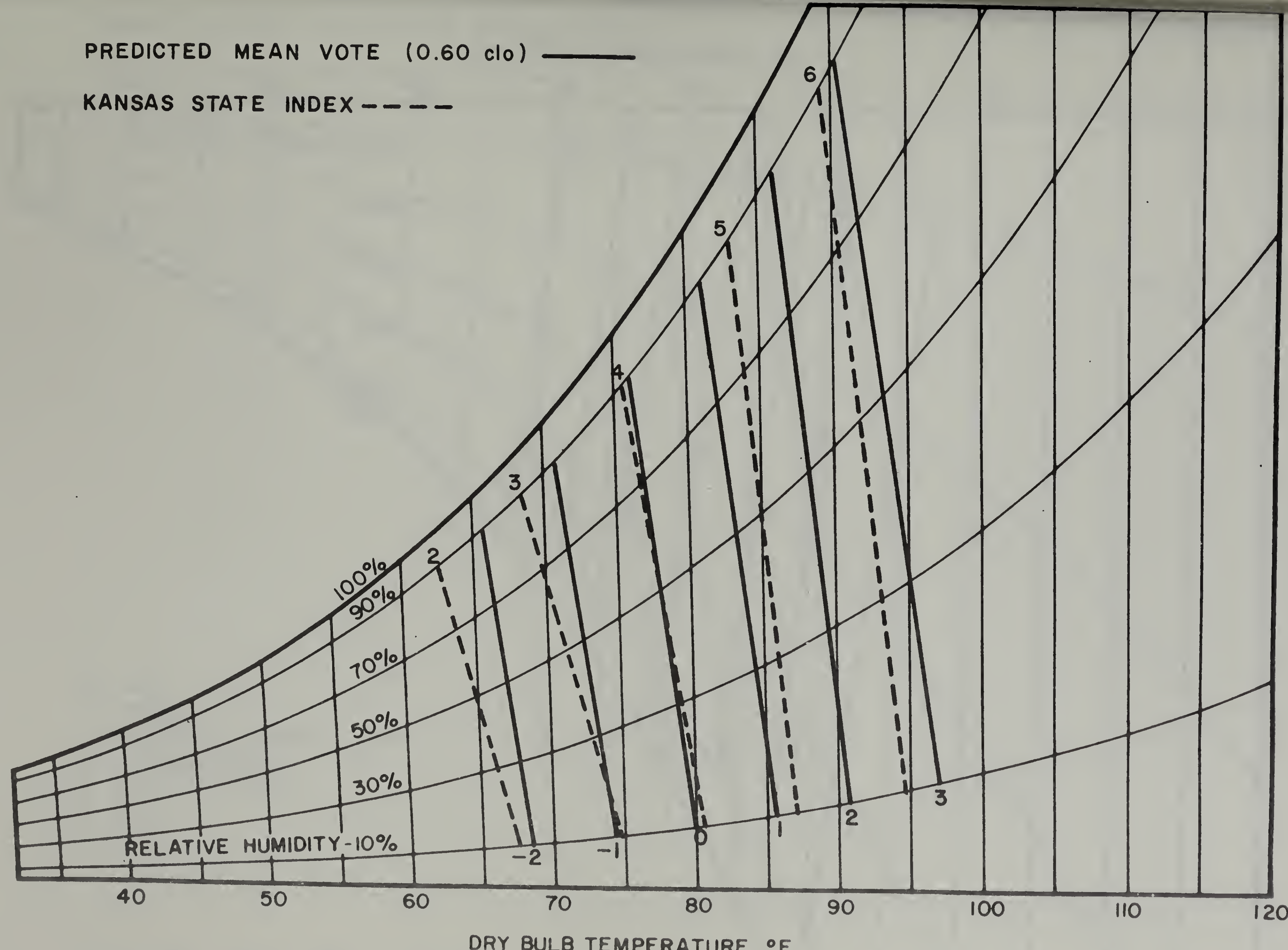

KANSAS STATE INDEX
MODEL COMFORT ENVELOPE
70%
60%
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
2
3
4
5
6
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 7



PREDICTED MEAN VOTE (0.60 clo)
−2 COOL
−1 SLIGHTLY COOL
0 COMFORTABLE
1 SLIGHTLY WARM
2 WARM
3 HOT
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
-2
-1
0
1
2
3
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 8



PREDICTED MEAN VOTE (0.60 clo)
KANSAS STATE INDEX
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
2
3
4
5
6
-2
-1
0
1
2
3
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 9

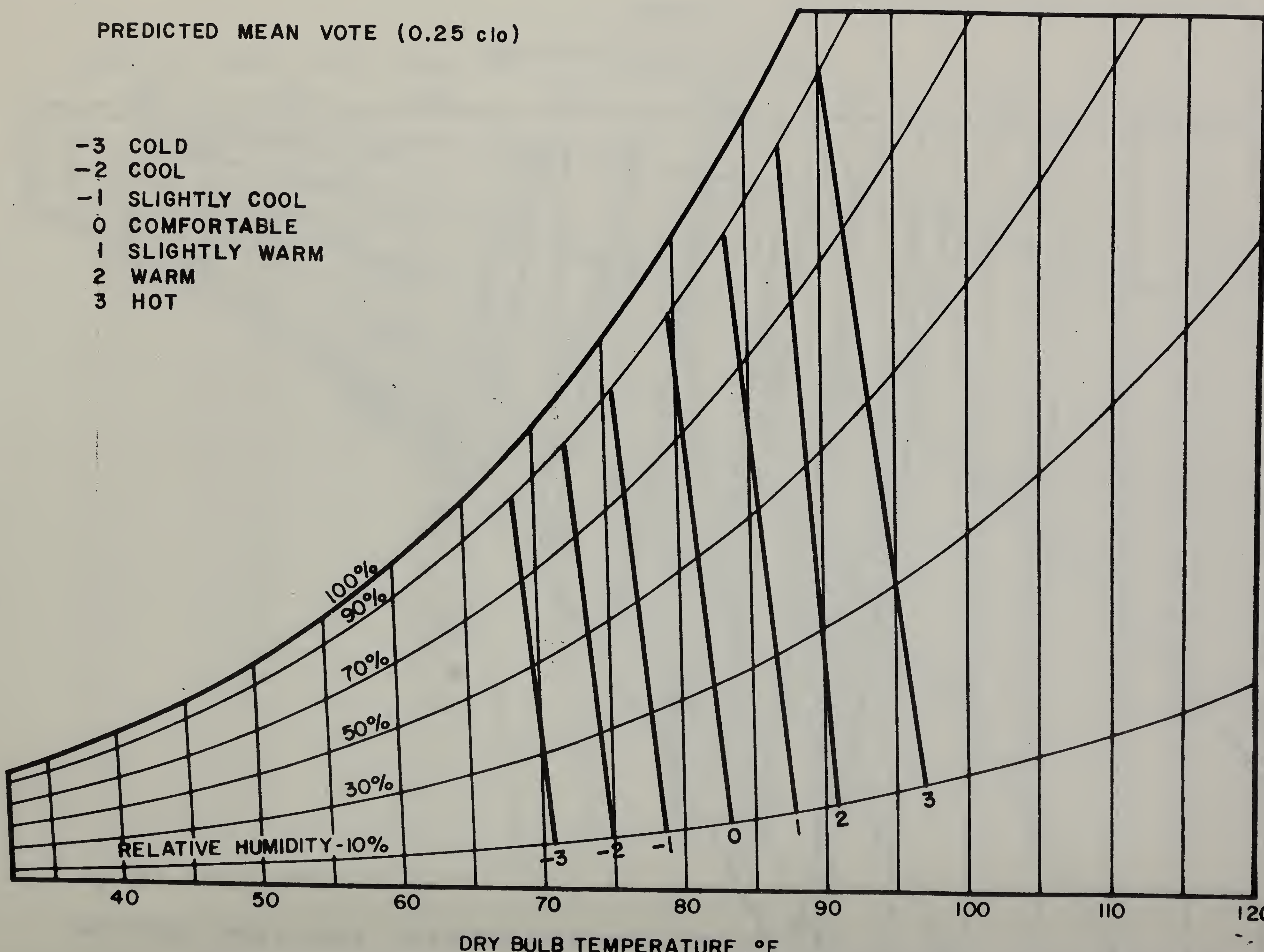

PREDICTED MEAN VOTE (0.25 clo)
-3 COLD
-2 COOL
-1 SLIGHTLY COOL
0 COMFORTABLE
1 SLIGHTLY WARM
2 WARM
3 HOT
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
-3
-2
-1
0
1
2
3
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 10

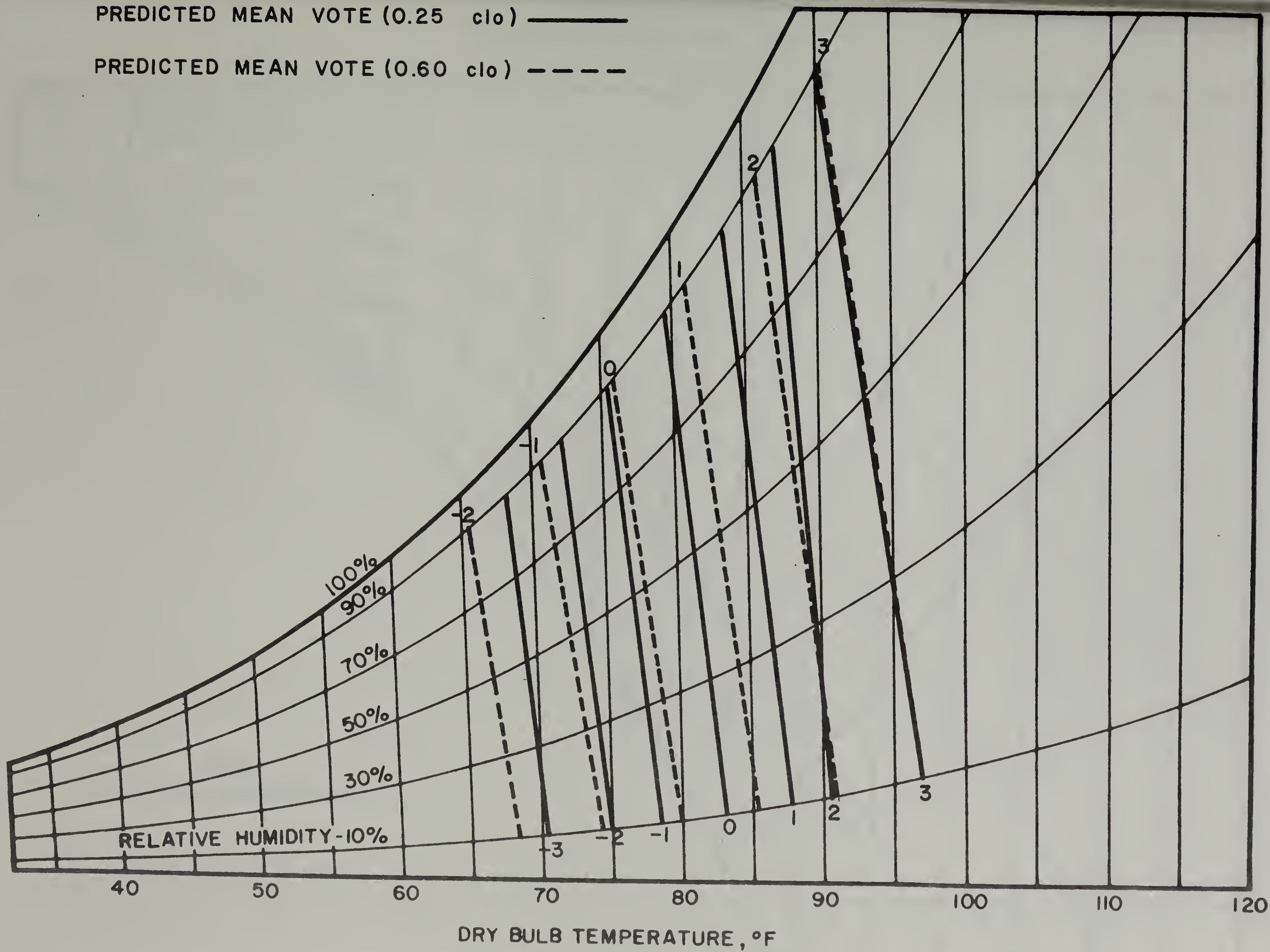

PREDICTED MEAN VOTE (0.25 clo)
PREDICTED MEAN VOTE (0.60 clo)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
-3
-2
-1
0
1
2
3
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 11

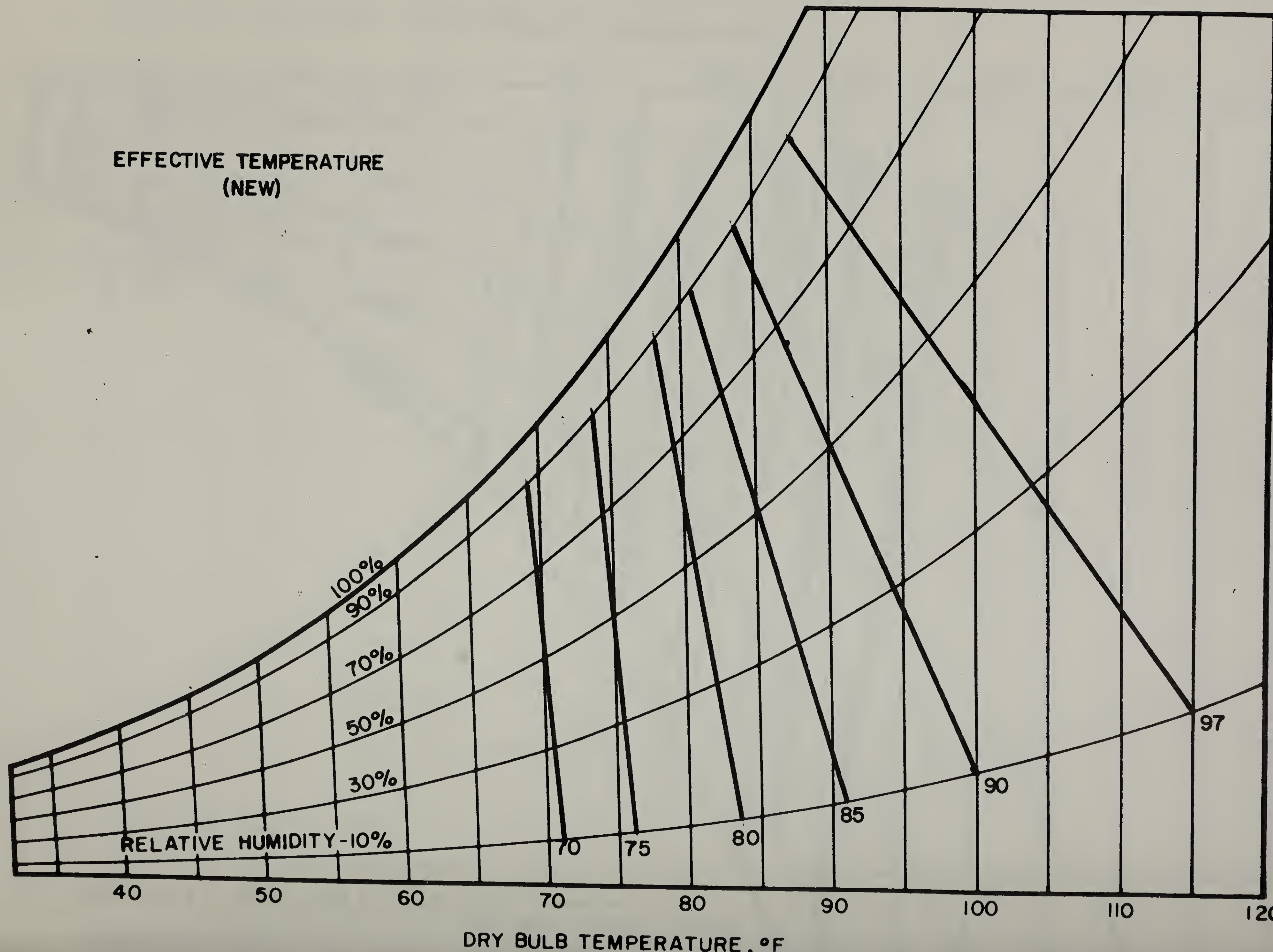

EFFECTIVE TEMPERATURE
(NEW)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
90
97
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 12

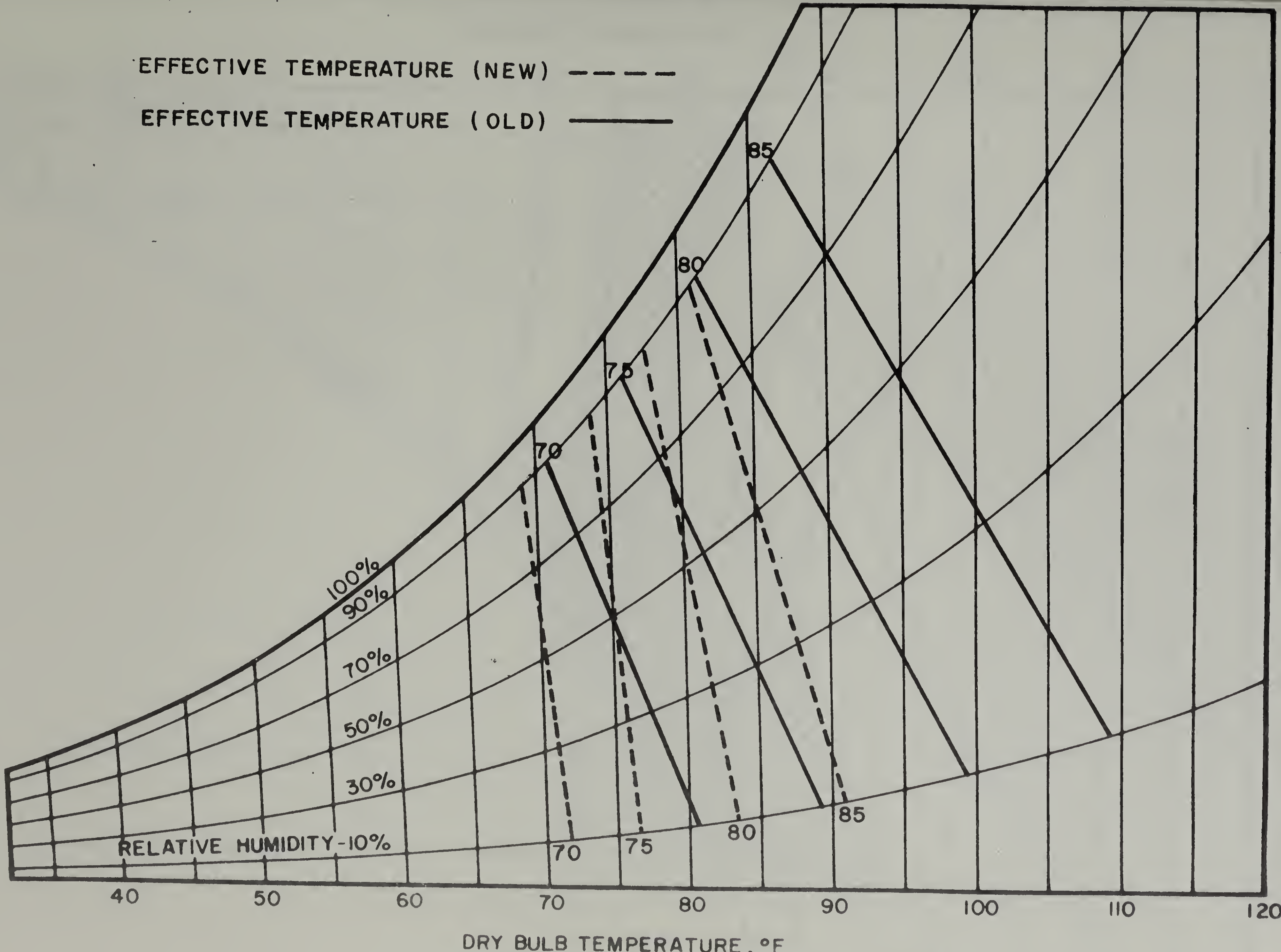

EFFECTIVE TEMPERATURE (NEW)
EFFECTIVE TEMPERATURE (OLD)
85
80
75
70
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 13

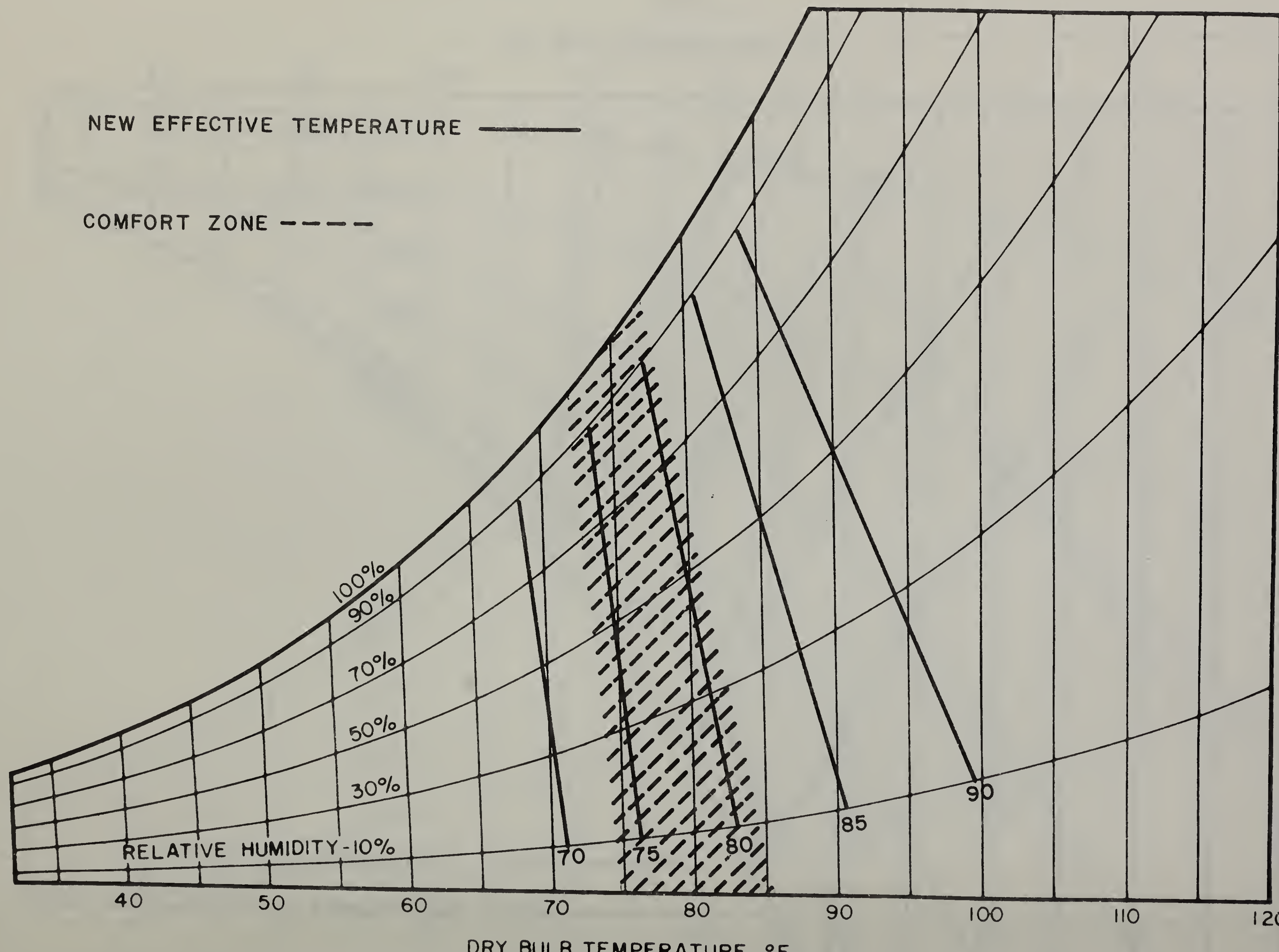

NEW EFFECTIVE TEMPERATURE
COMFORT ZONE
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 14

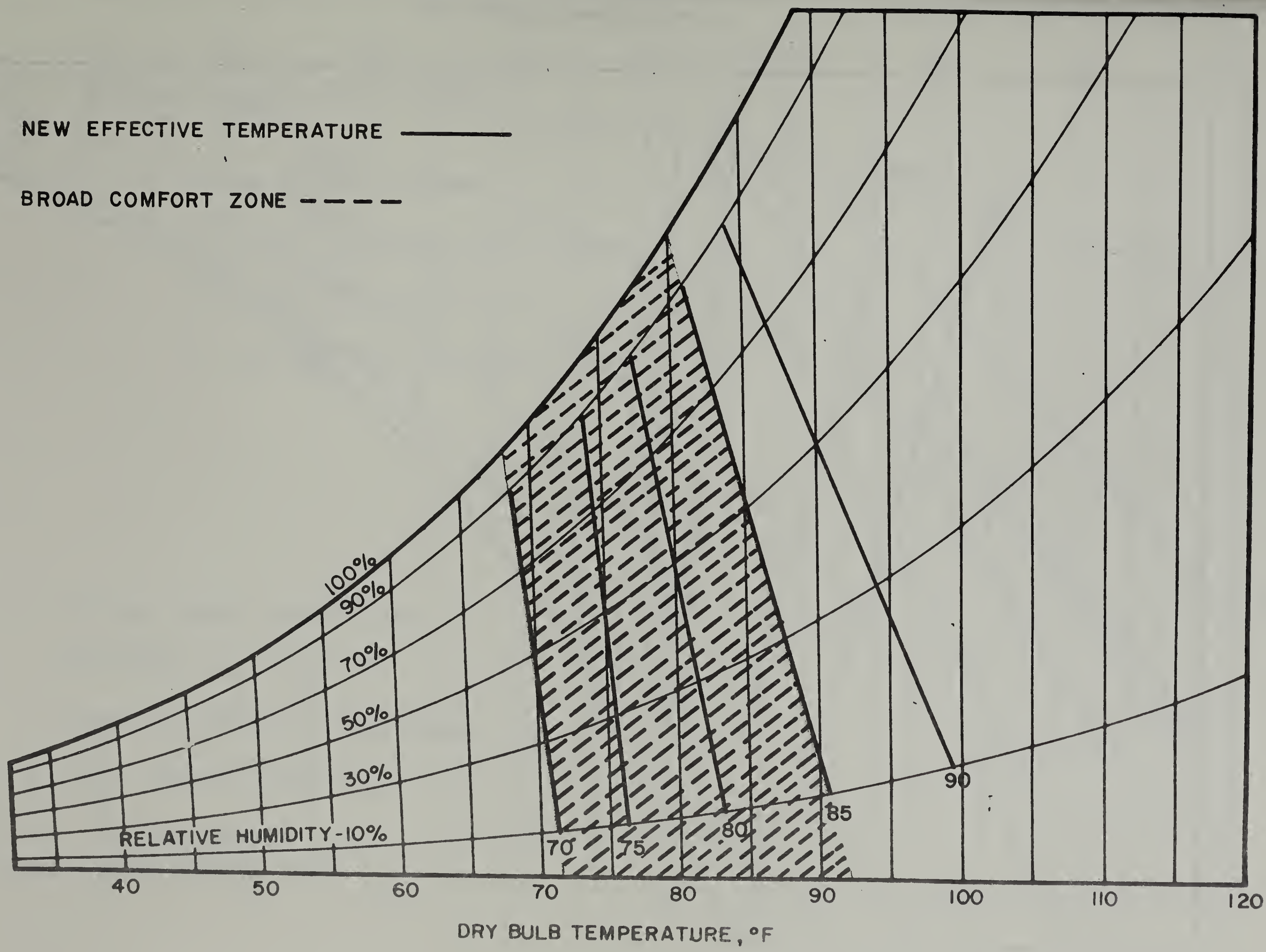

NEW EFFECTIVE TEMPERATURE
BROAD COMFORT ZONE
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 15

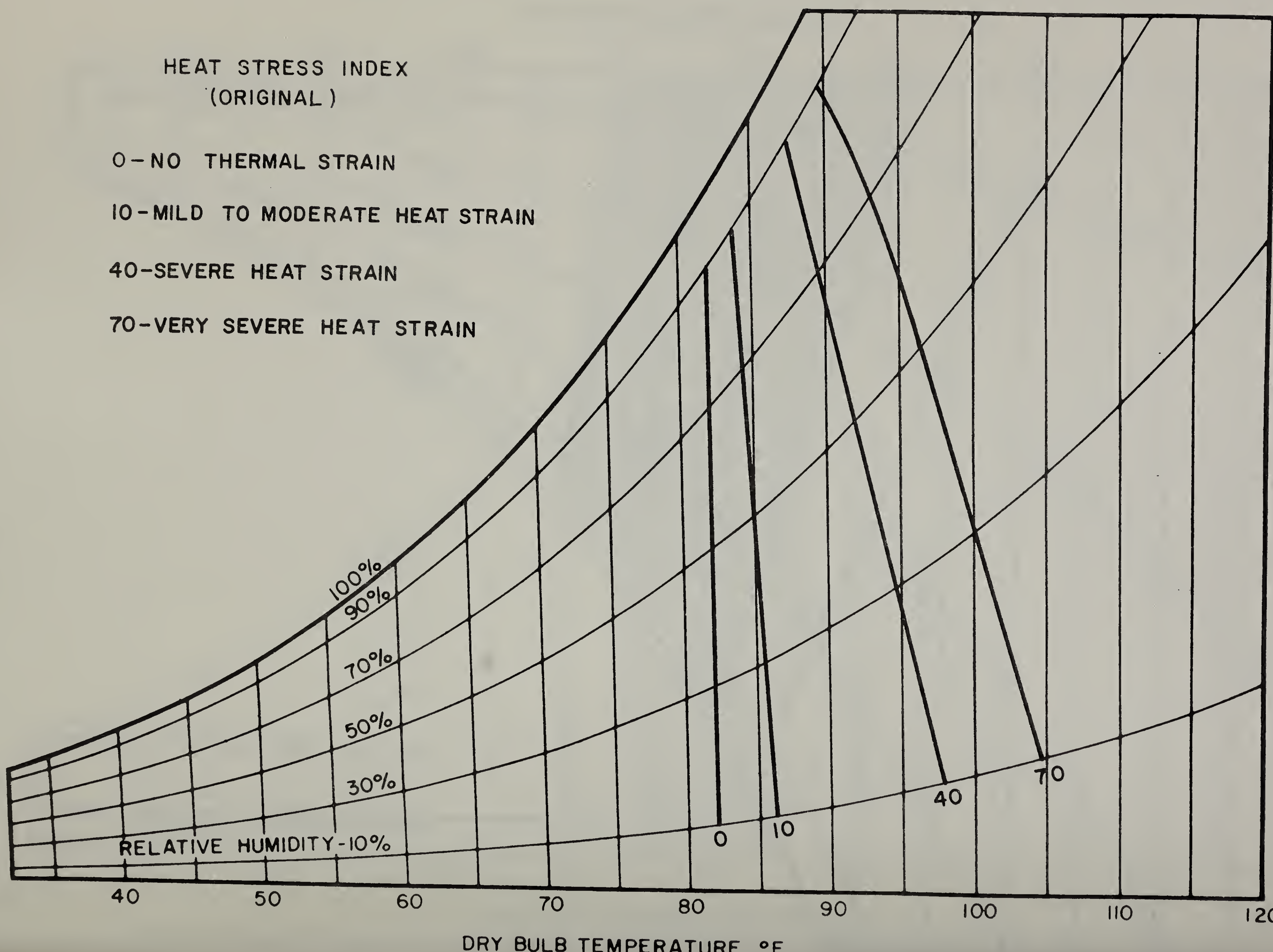

HEAT STRESS INDEX
(ORIGINAL)
0-NO THERMAL STRAIN
10-MILD TO MODERATE HEAT STRAIN
40-SEVERE HEAT STRAIN
70-VERY SEVERE HEAT STRAIN
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 16

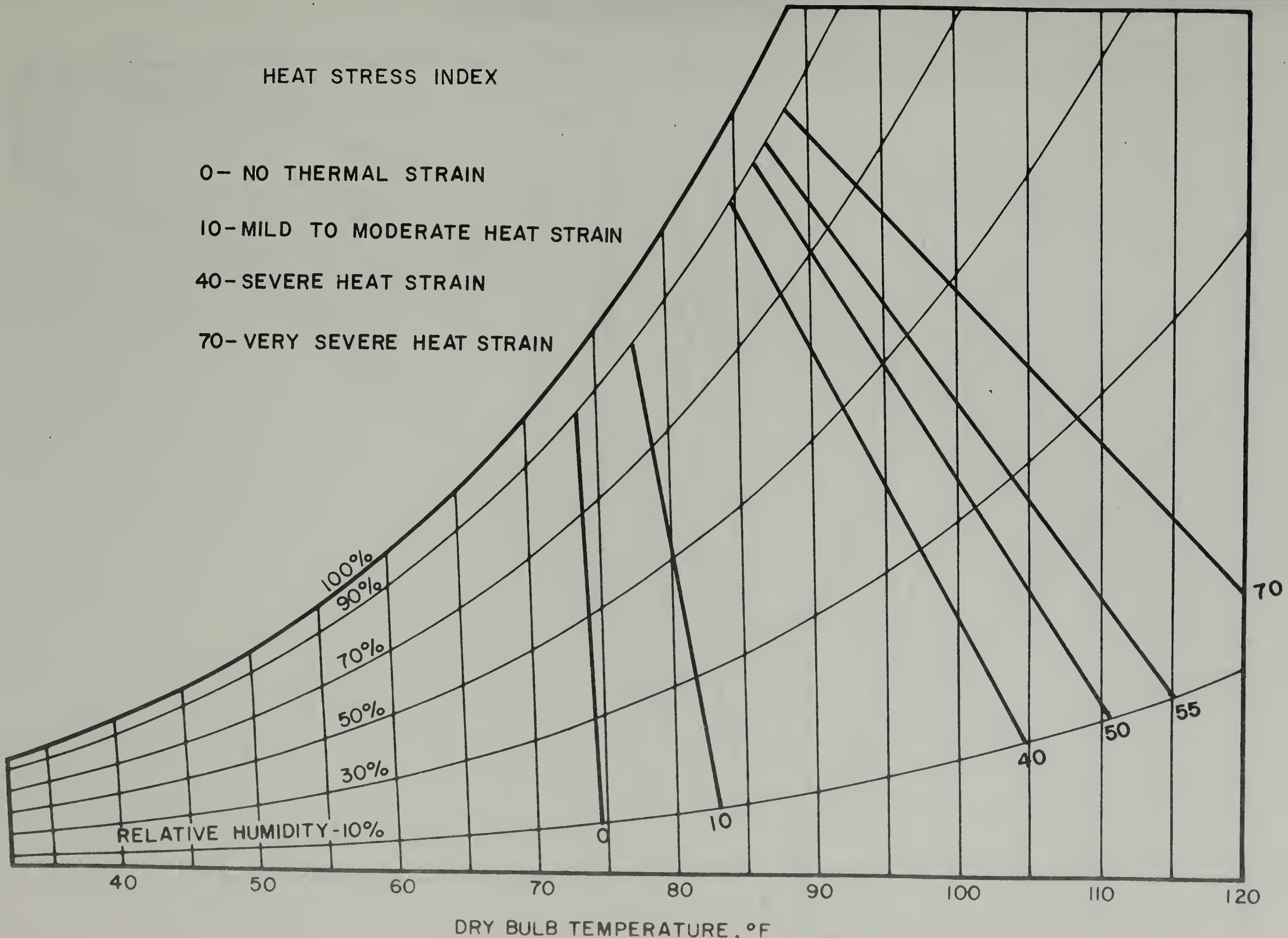
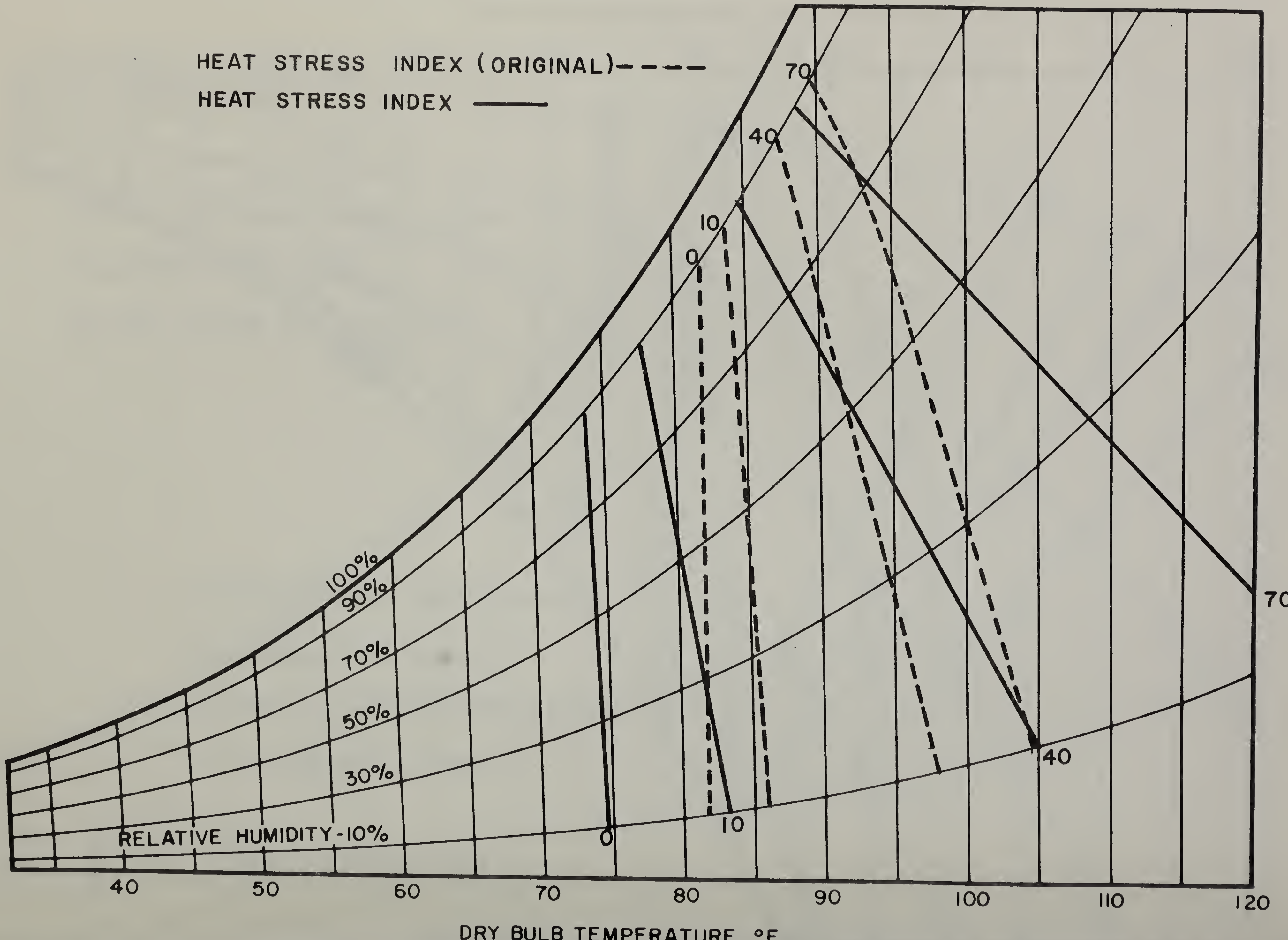

HEAT STRESS INDEX
0- NO THERMAL STRAIN
10-MILD TO MODERATE HEAT STRAIN
40-SEVERE HEAT STRAIN
70-VERY SEVERE HEAT STRAIN
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
50
55
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 17



HEAT STRESS INDEX (ORIGINAL)----
HEAT STRESS INDEX
70
40
10
0
70
40
10
0
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 18

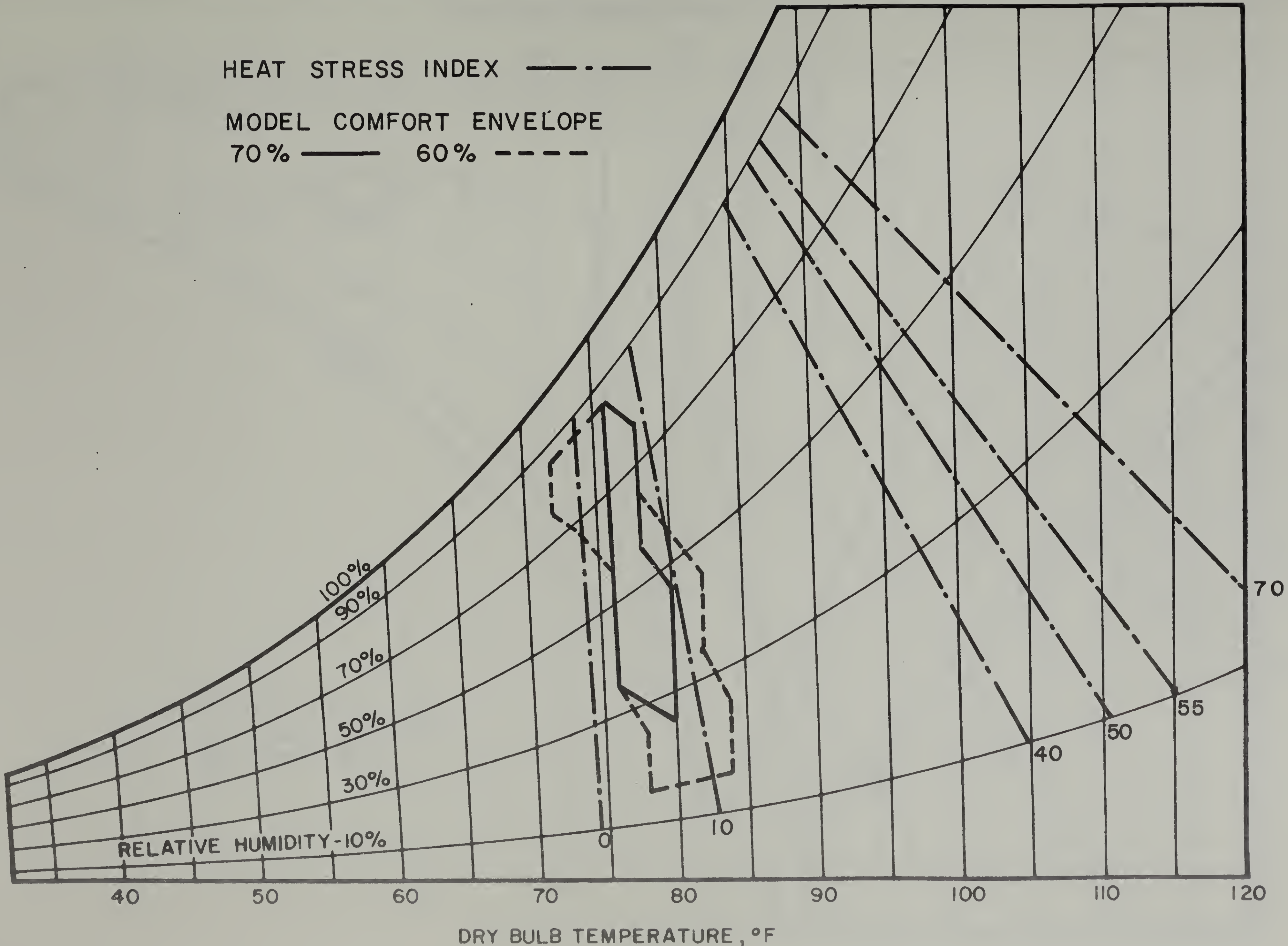
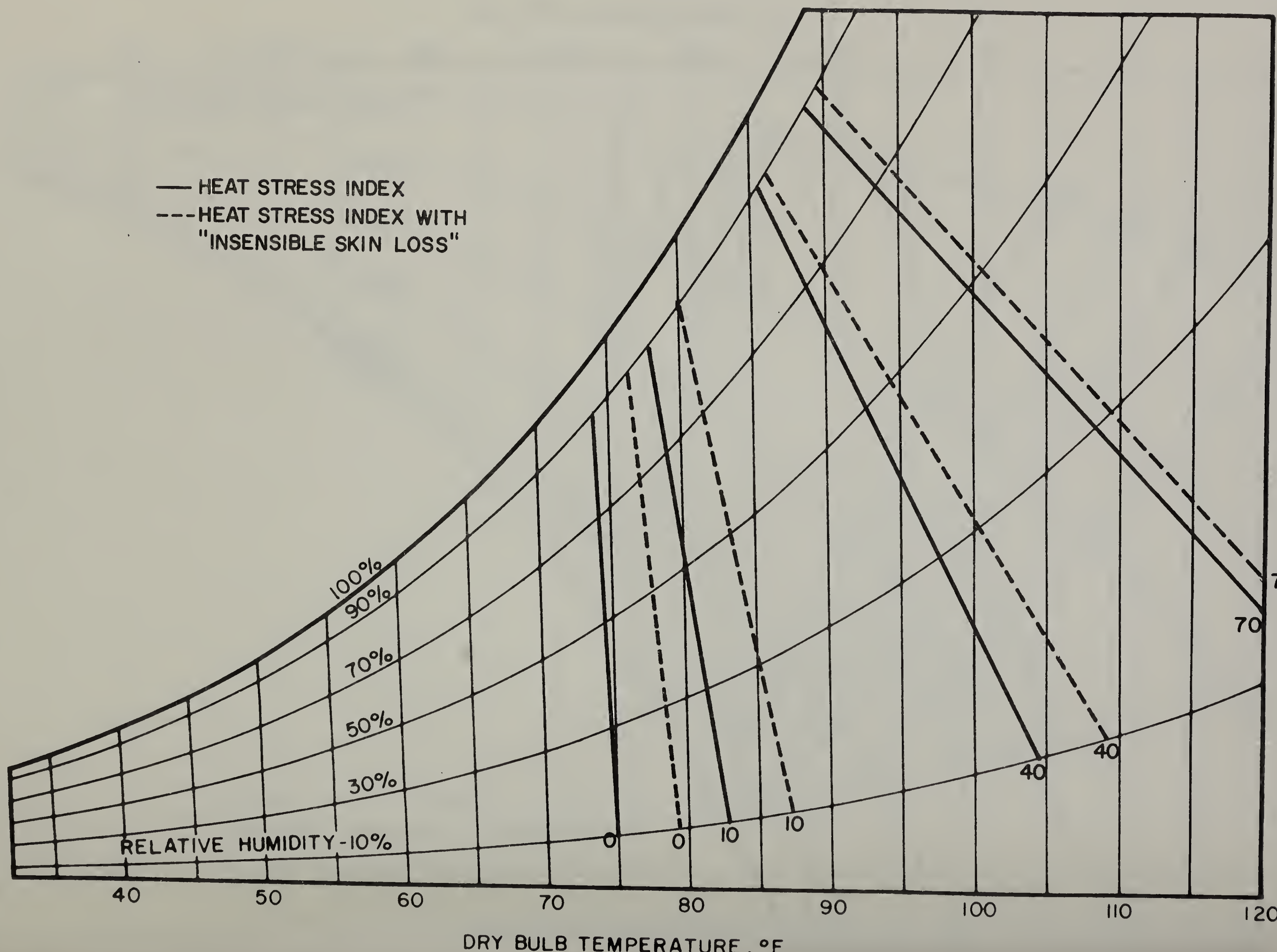

HEAT STRESS INDEX
MODEL COMFORT ENVELOPE
70%
60%
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
50
55
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 19



— HEAT STRESS INDEX
--- HEAT STRESS INDEX WITH "INSENSIBLE SKIN LOSS"
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
0
10
10
40
40
70
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 20

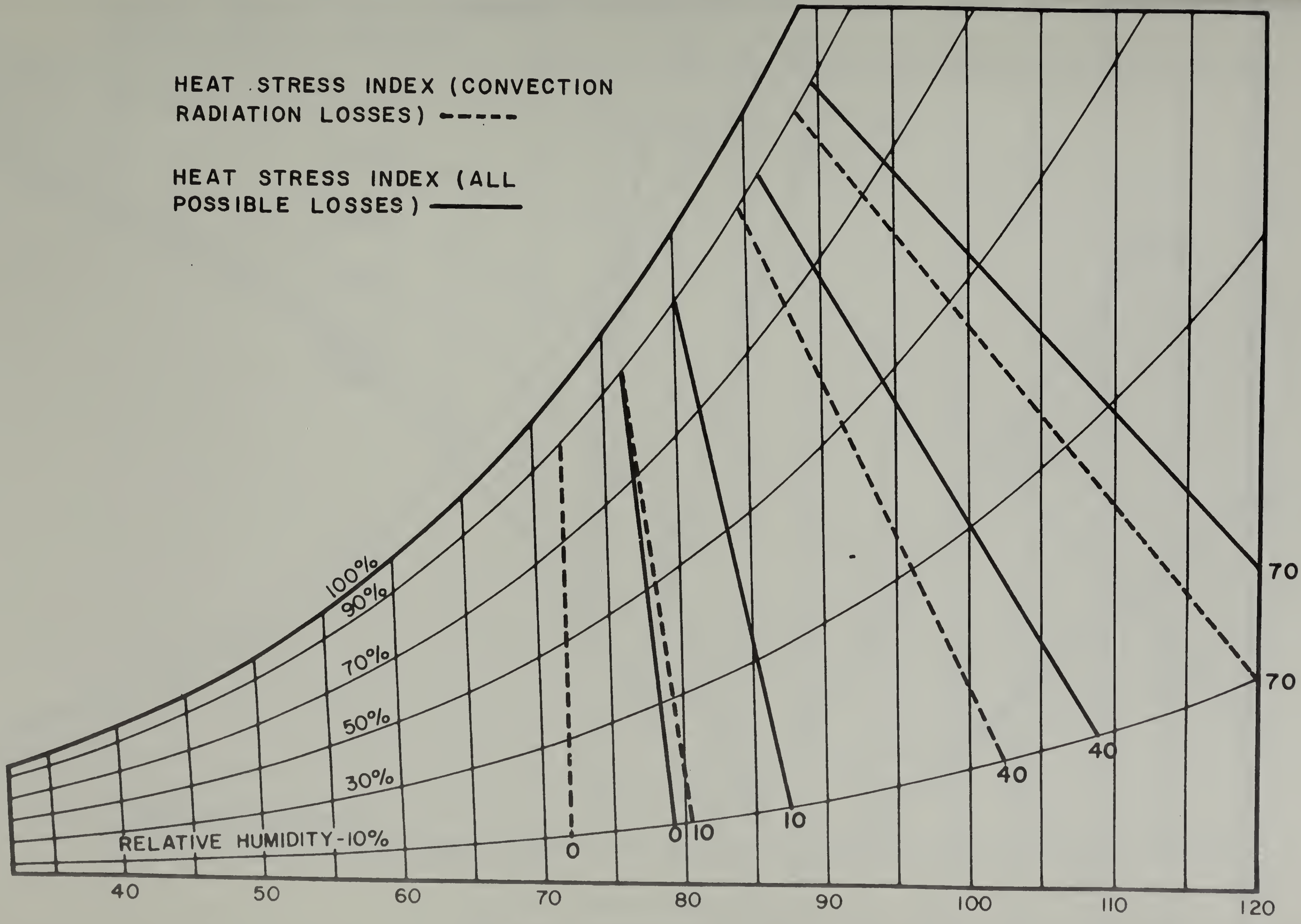
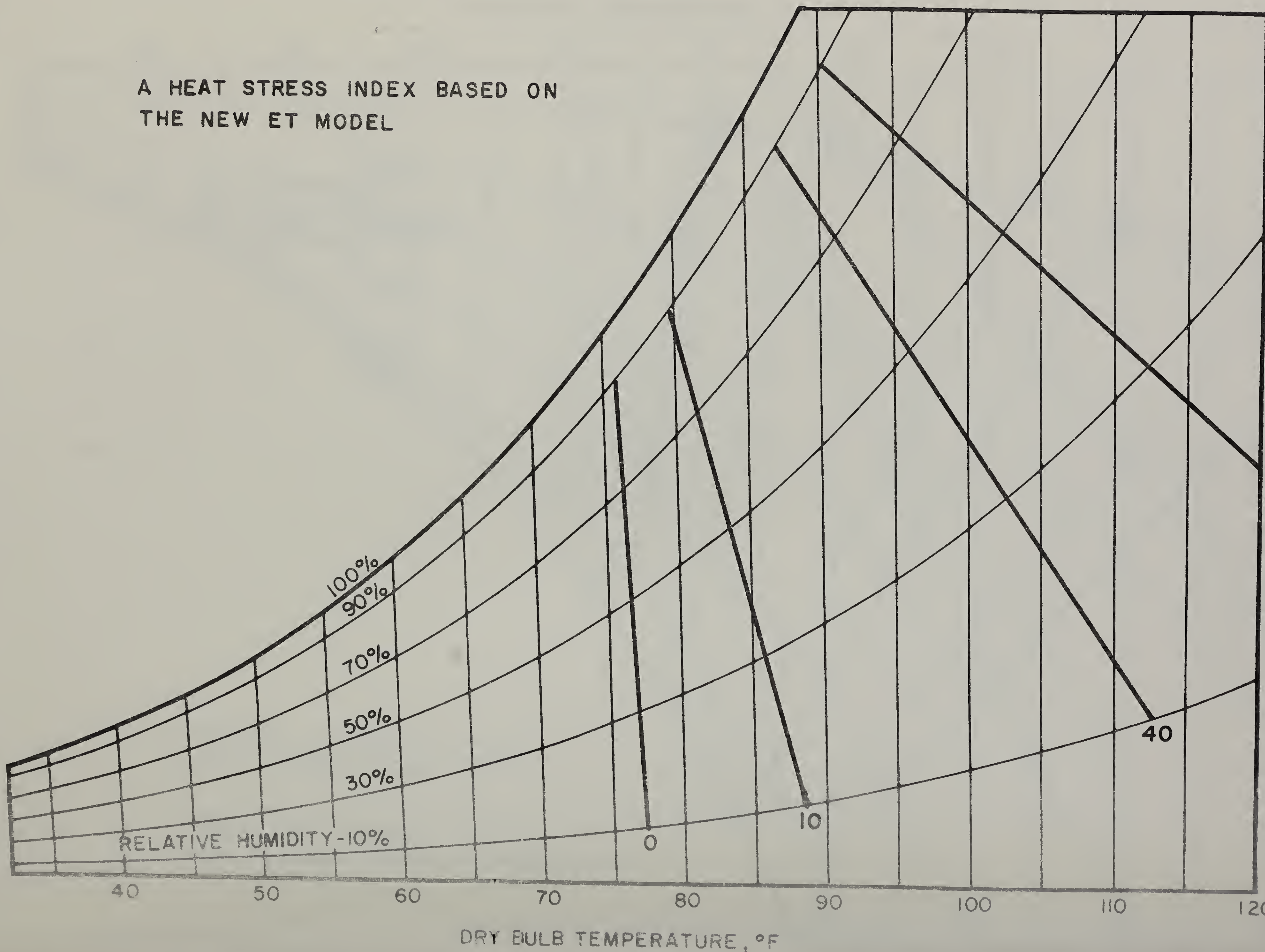

HEAT STRESS INDEX (CONVECTION RADIATION LOSSES)
HEAT STRESS INDEX (ALL POSSIBLE LOSSES)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
0
10
10
40
40
70
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 21



A HEAT STRESS INDEX BASED ON
THE NEW ET MODEL
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 22

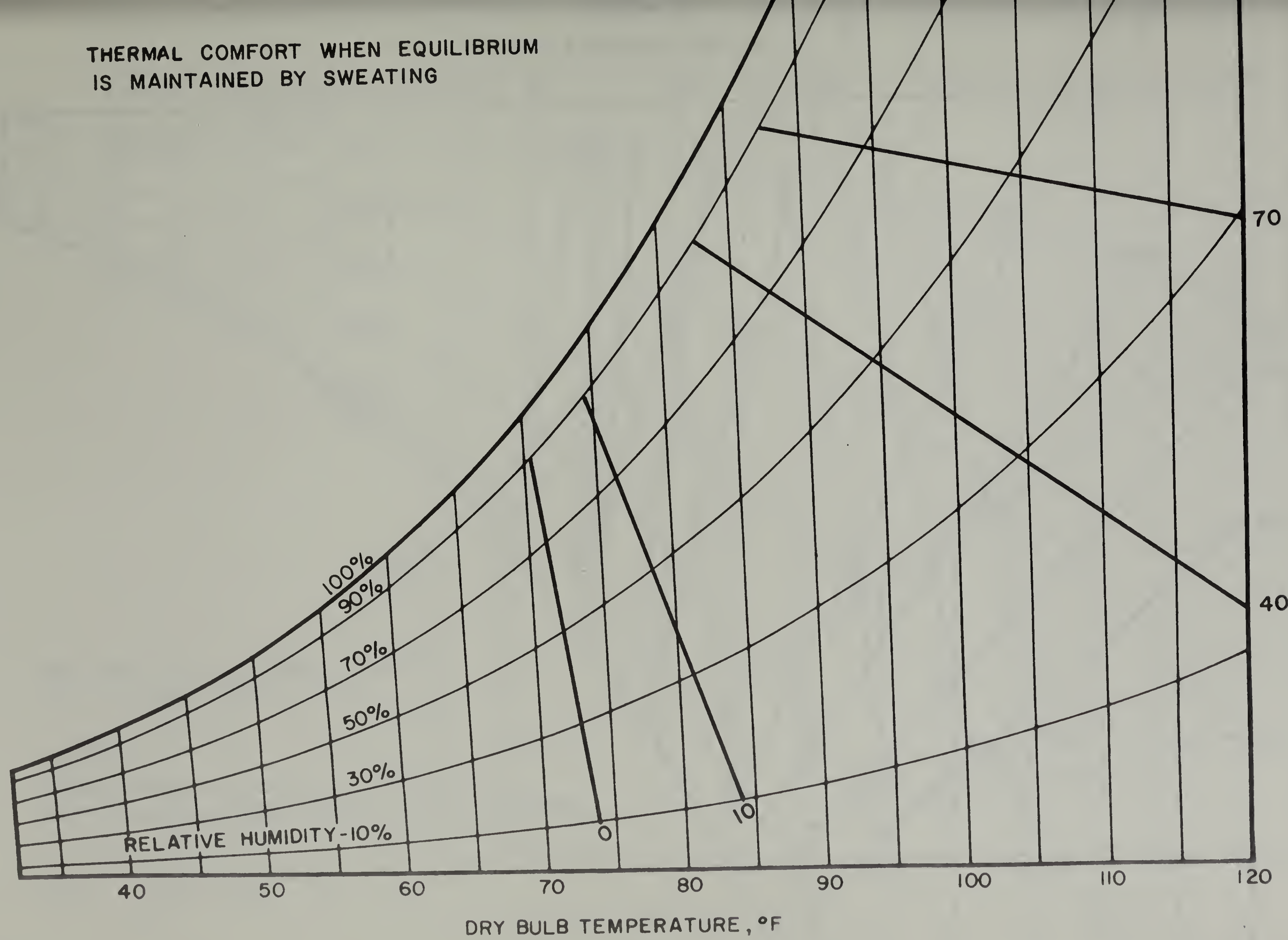

THERMAL COMFORT WHEN EQUILIBRIUM
IS MAINTAINED BY SWEATING
70
40
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 23

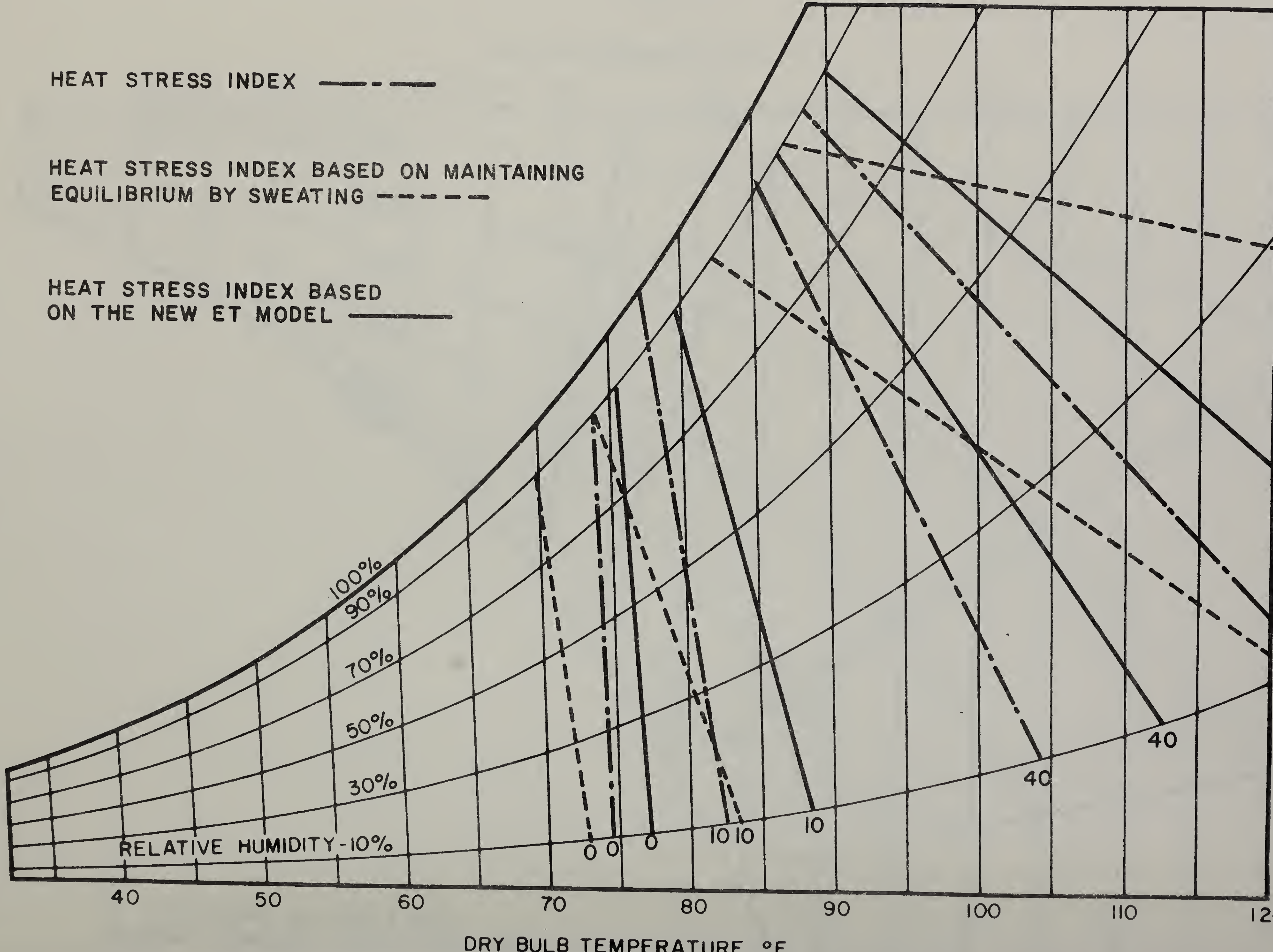

HEAT STRESS INDEX
HEAT STRESS INDEX BASED ON MAINTAINING EQUILIBRIUM BY SWEATING
HEAT STRESS INDEX BASED ON THE NEW ET MODEL
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
0
0
10
10
10
40
40
70
70
70
40
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 24

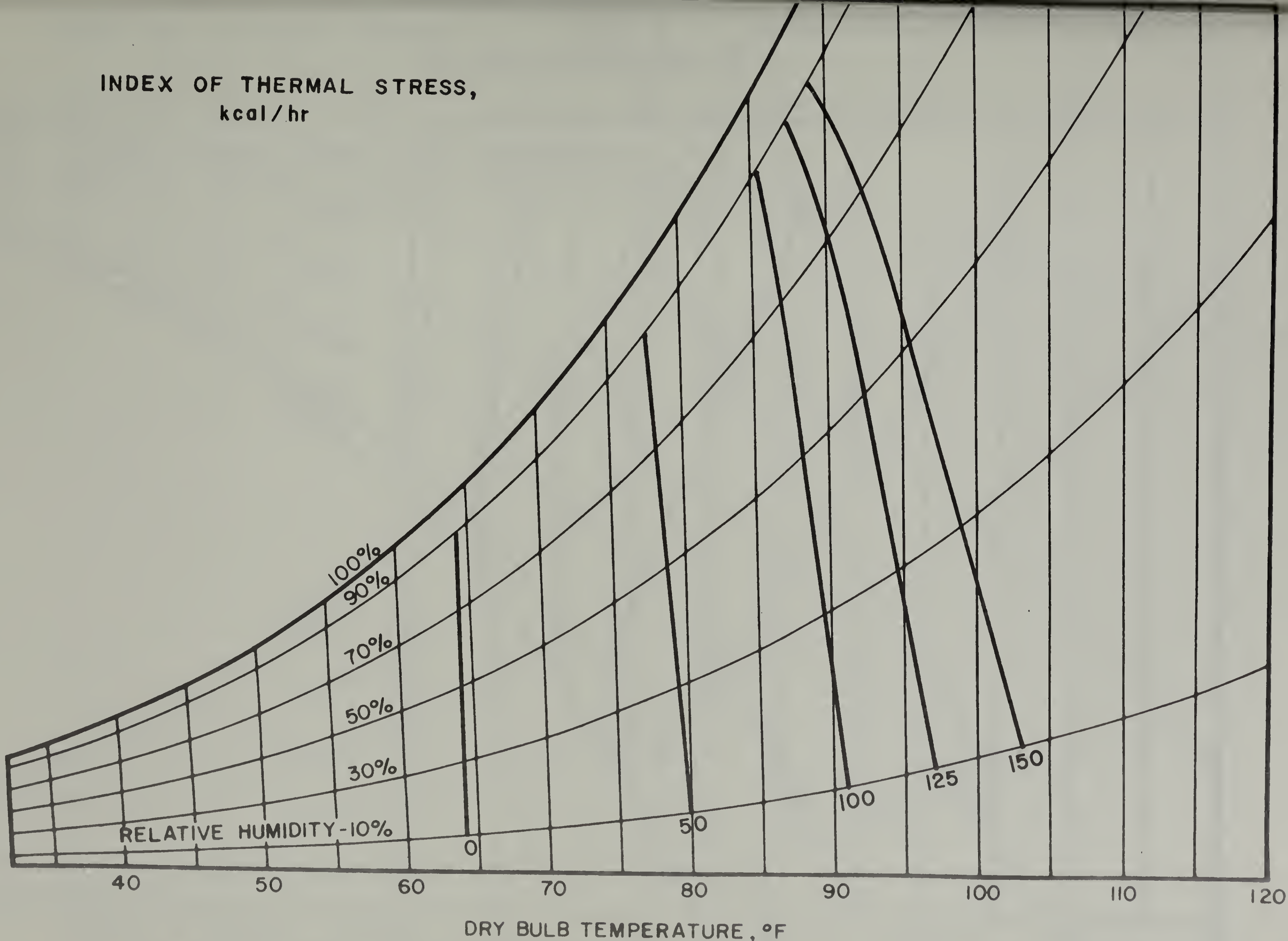

INDEX OF THERMAL STRESS,
kcal/hr
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
50
100
125
150
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 25

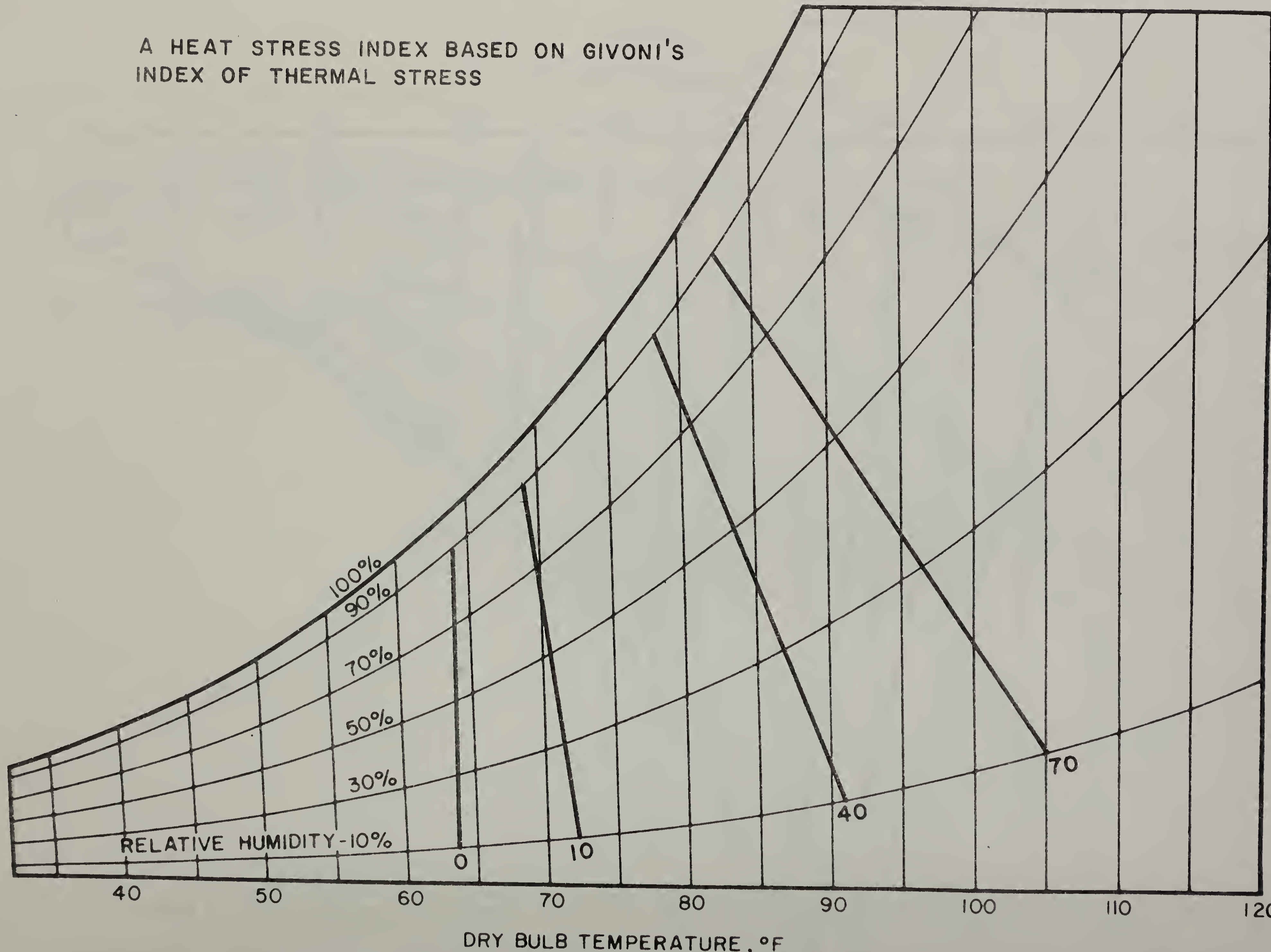

A HEAT STRESS INDEX BASED ON GIVONI'S INDEX OF THERMAL STRESS
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
10
40
70
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 26

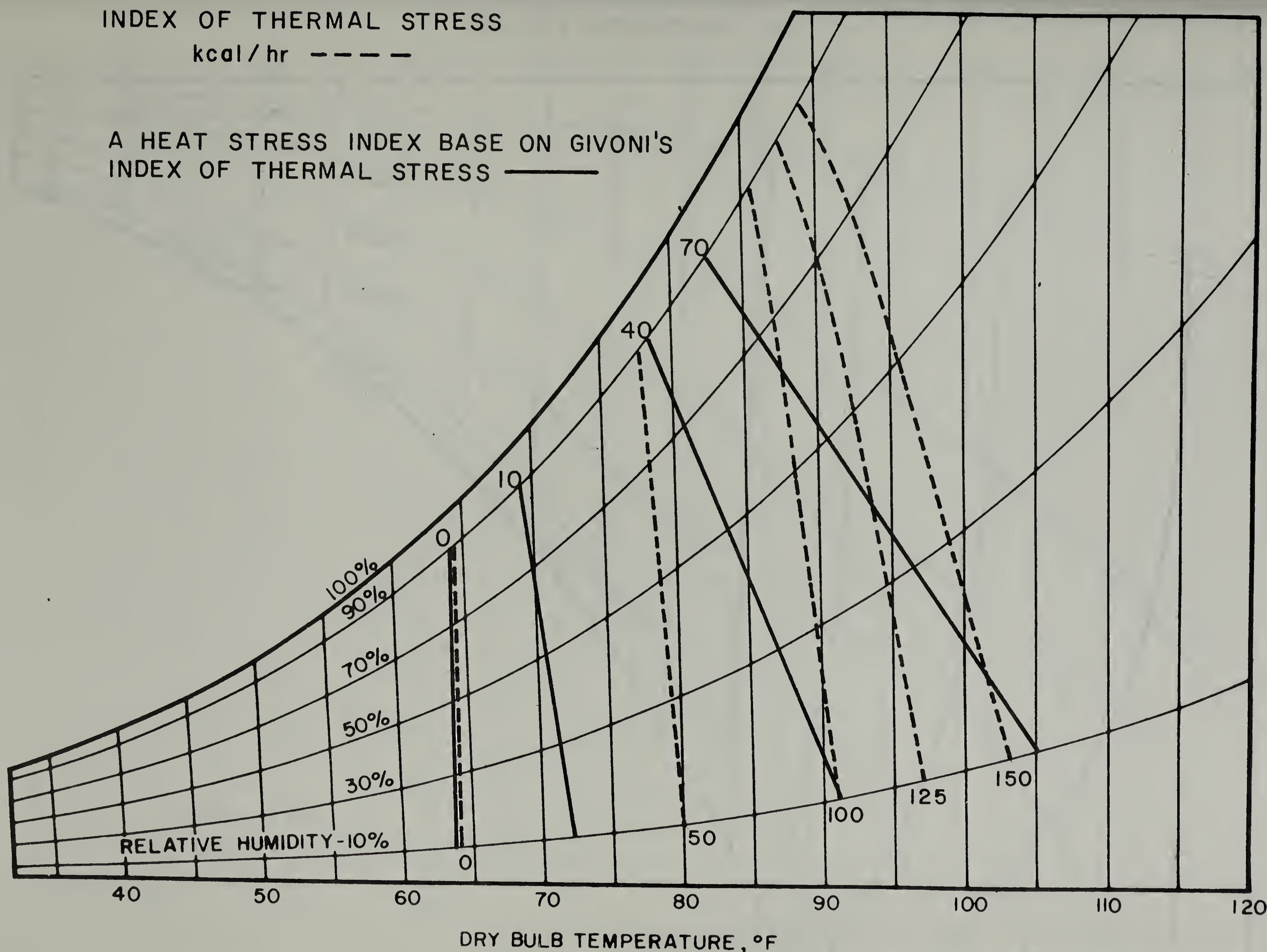

INDEX OF THERMAL STRESS
kcal/hr ----
A HEAT STRESS INDEX BASE ON GIVONI'S
INDEX OF THERMAL STRESS ——
0
10
40
70
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
50
100
125
150
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 27

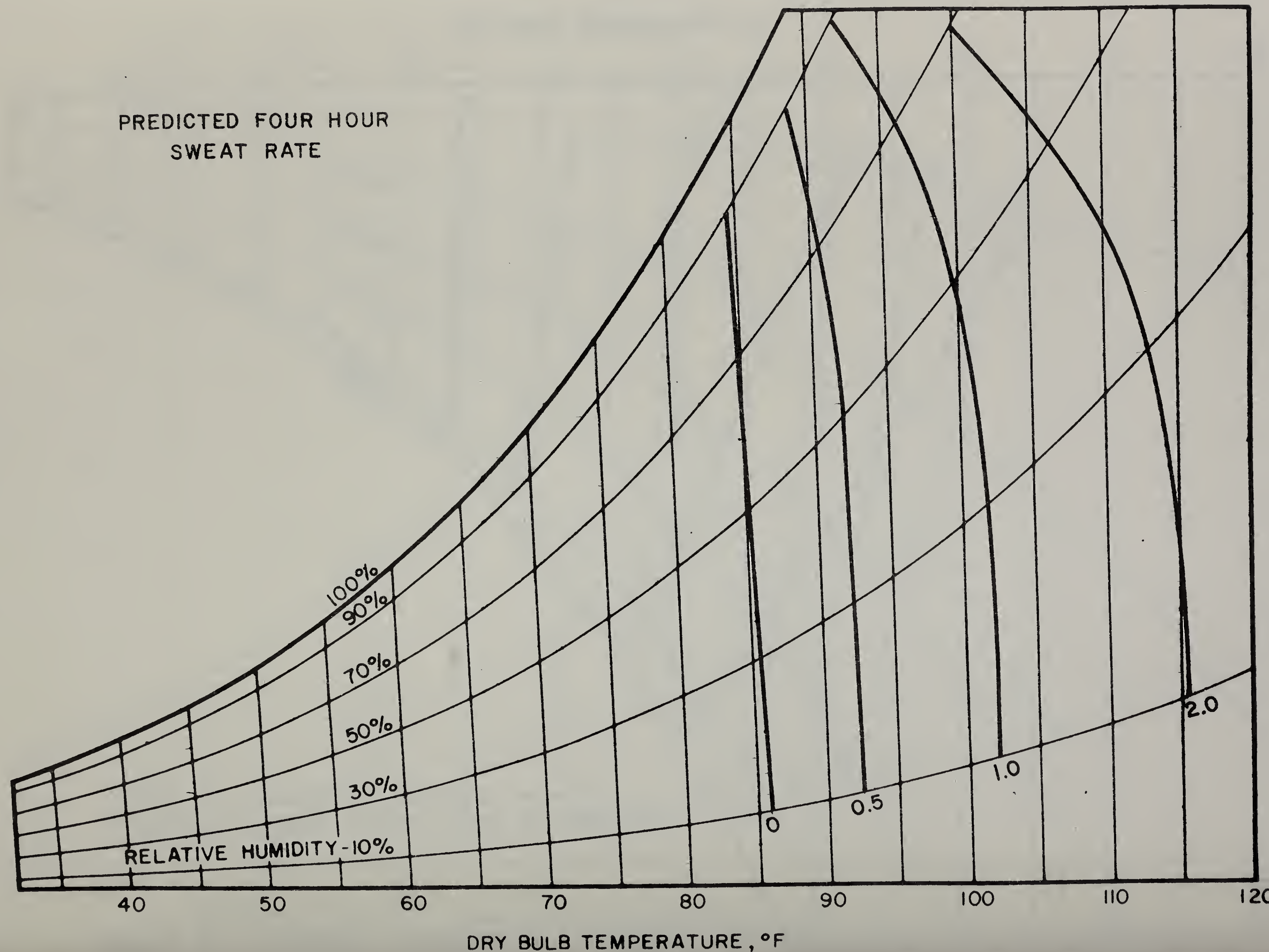
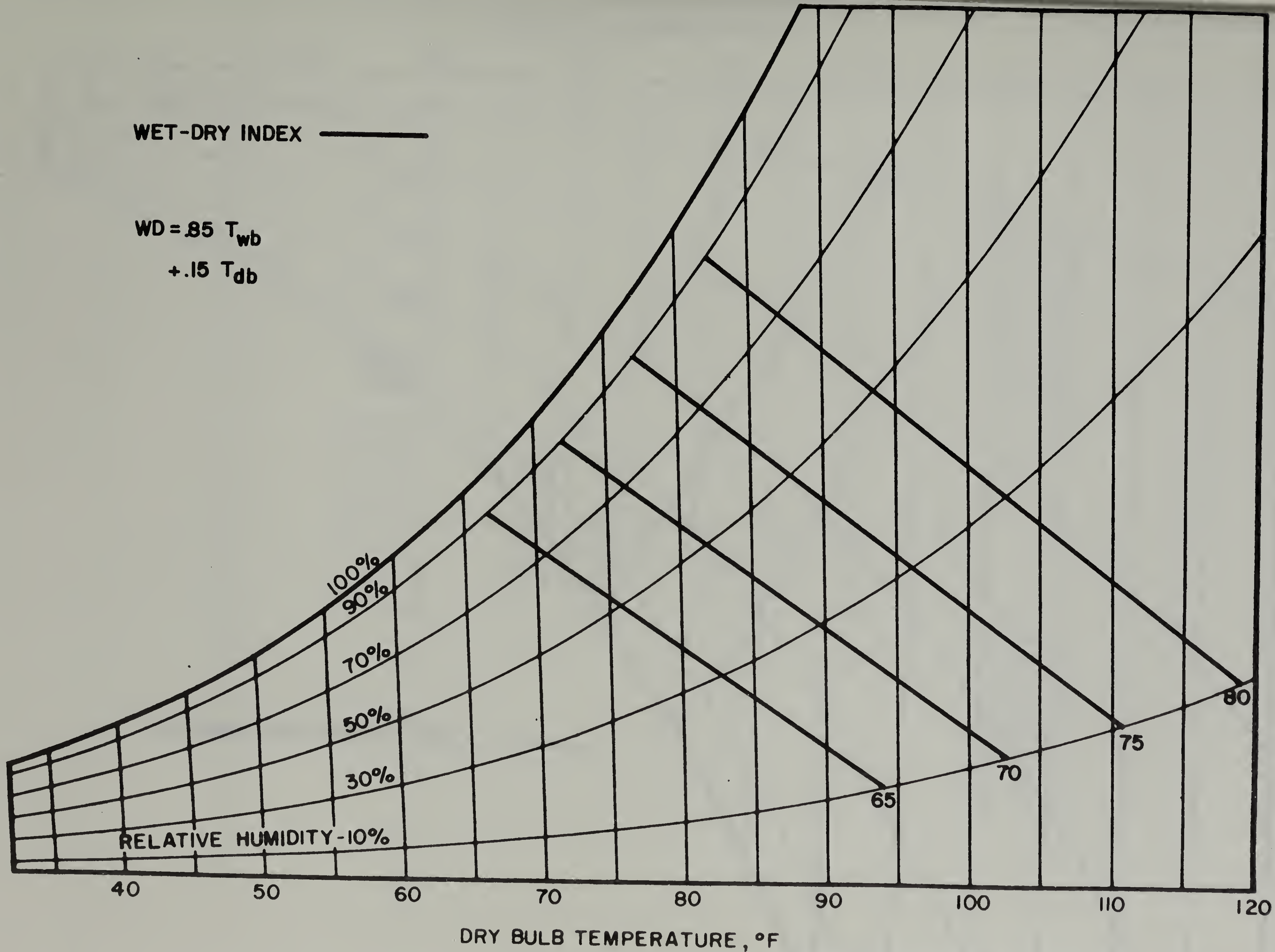
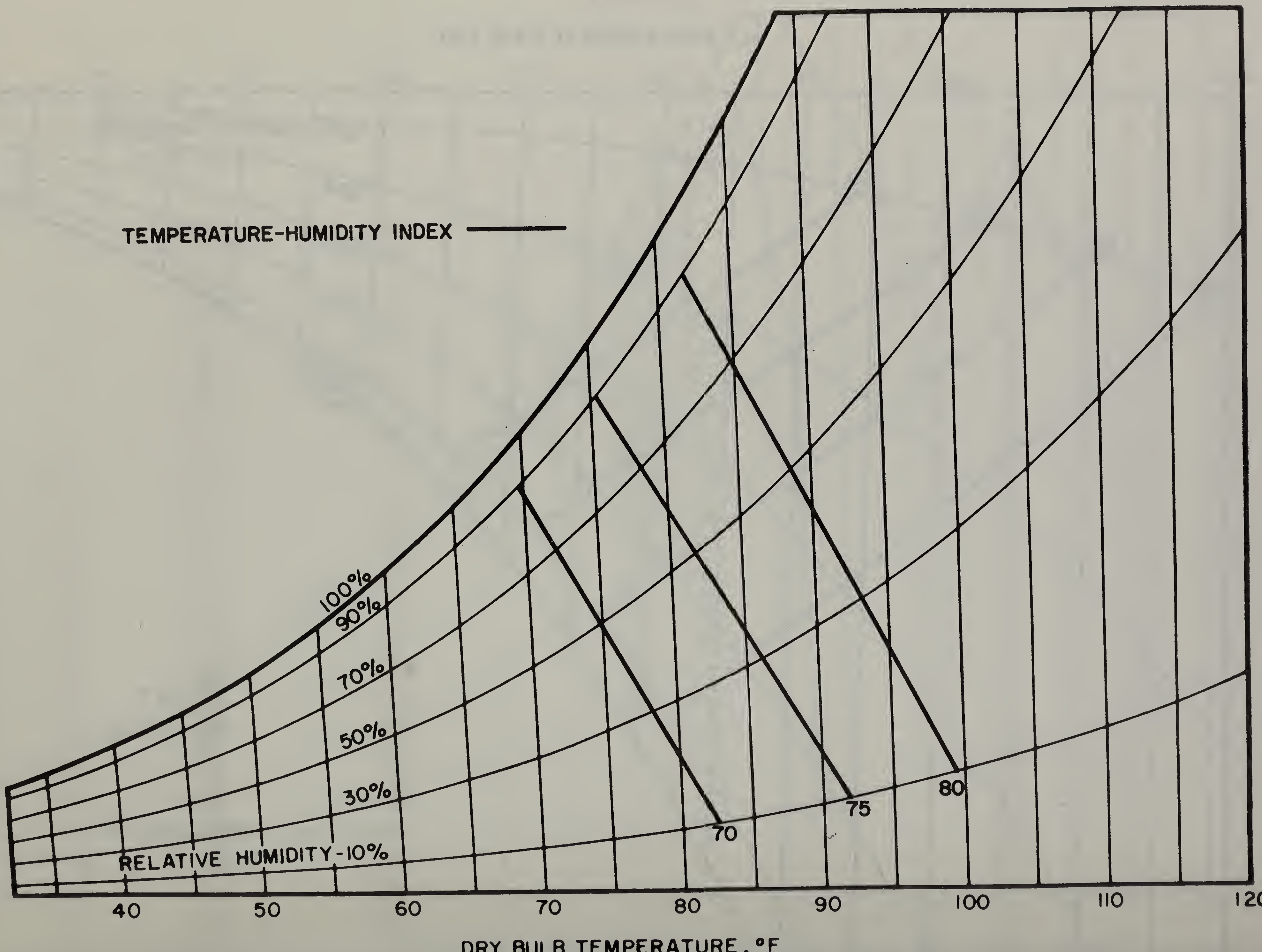

PREDICTED FOUR HOUR
SWEAT RATE
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
0
0.5
1.0
2.0
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 28



WET-DRY INDEX
WD = .85 T_{wb}
+.15 T_{db}
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
65
70
75
80
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

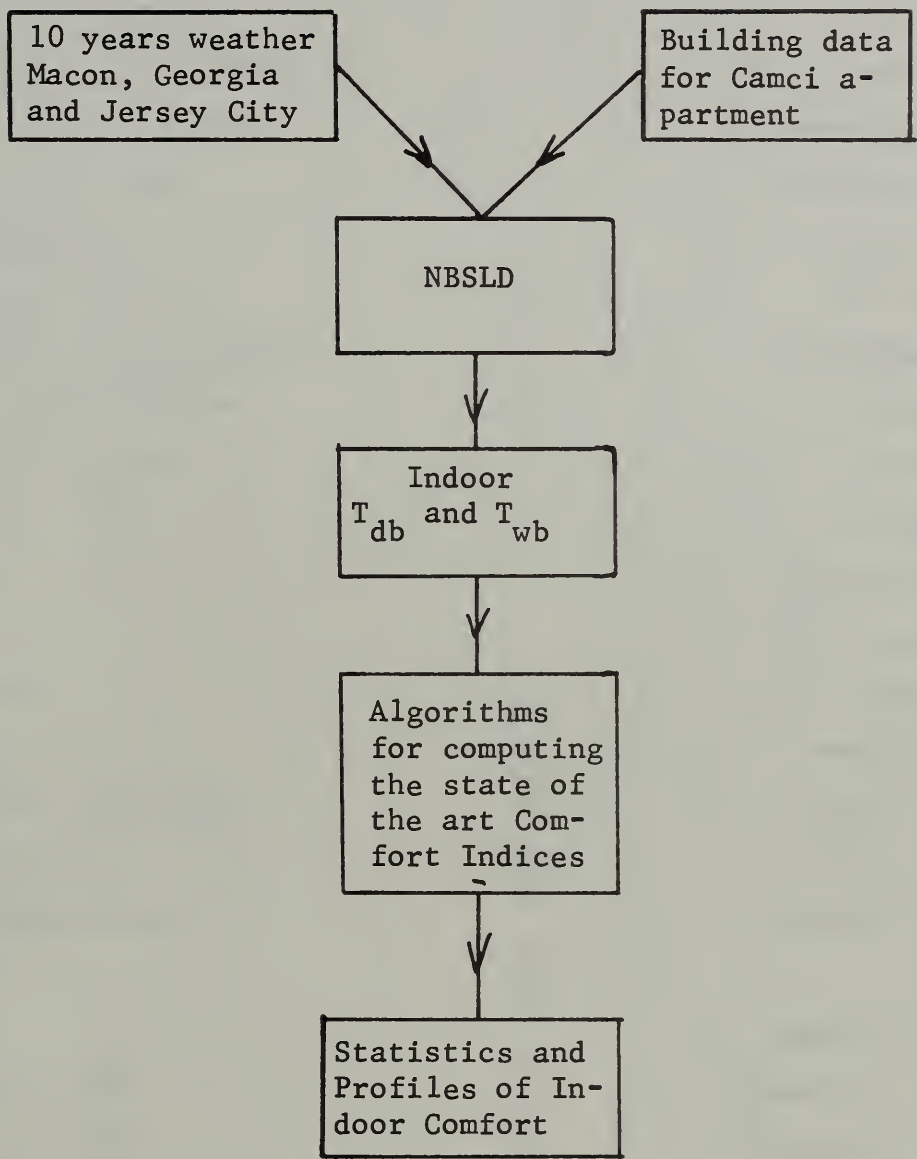
Figure 29



TEMPERATURE-HUMIDITY INDEX
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 30

Feasibility Study



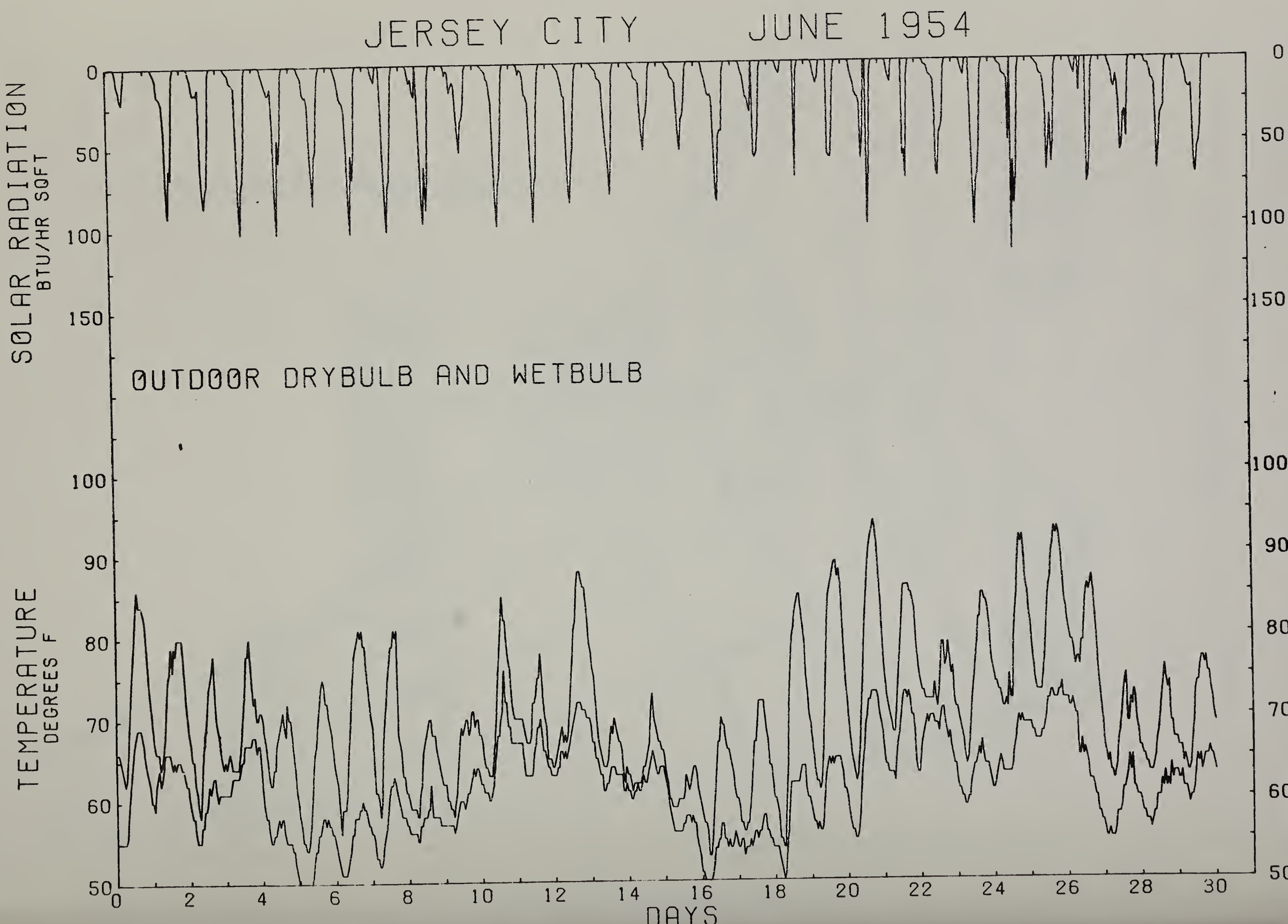
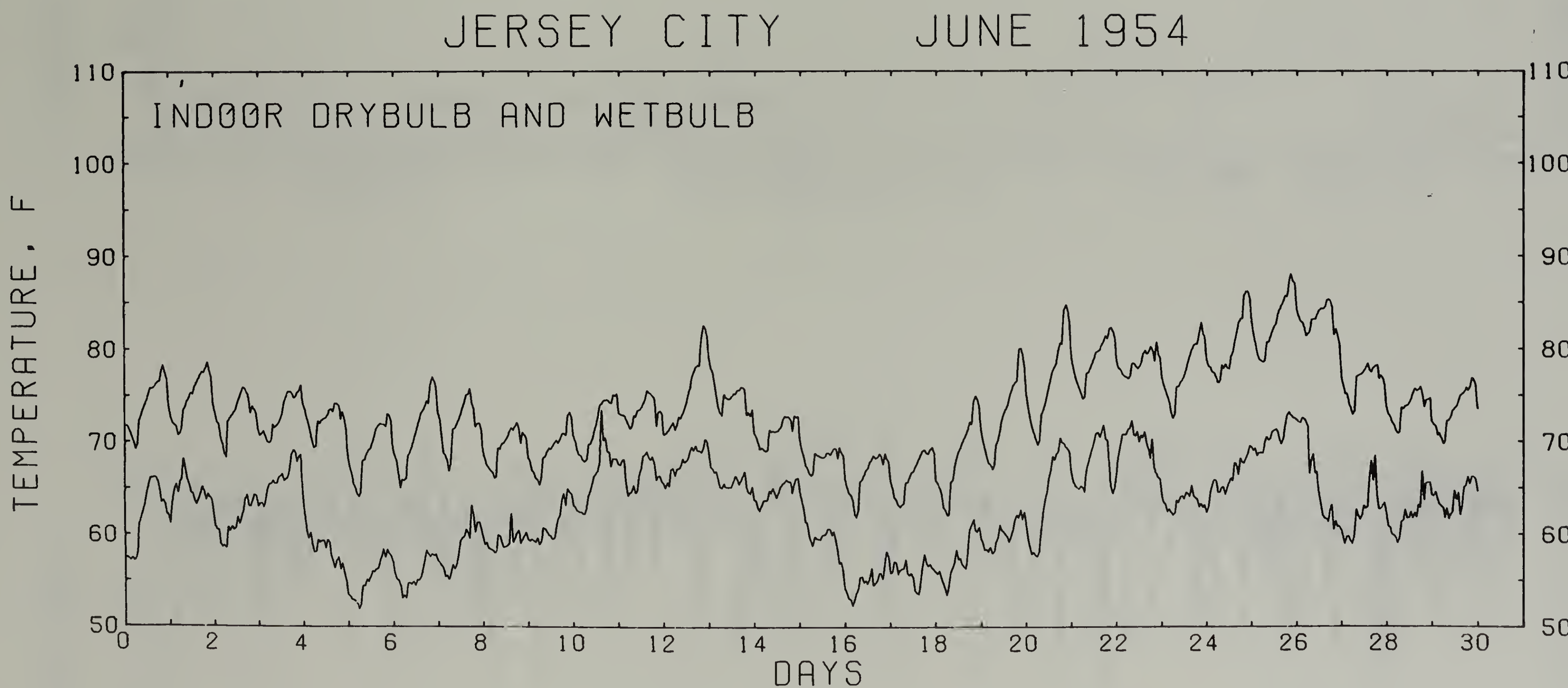
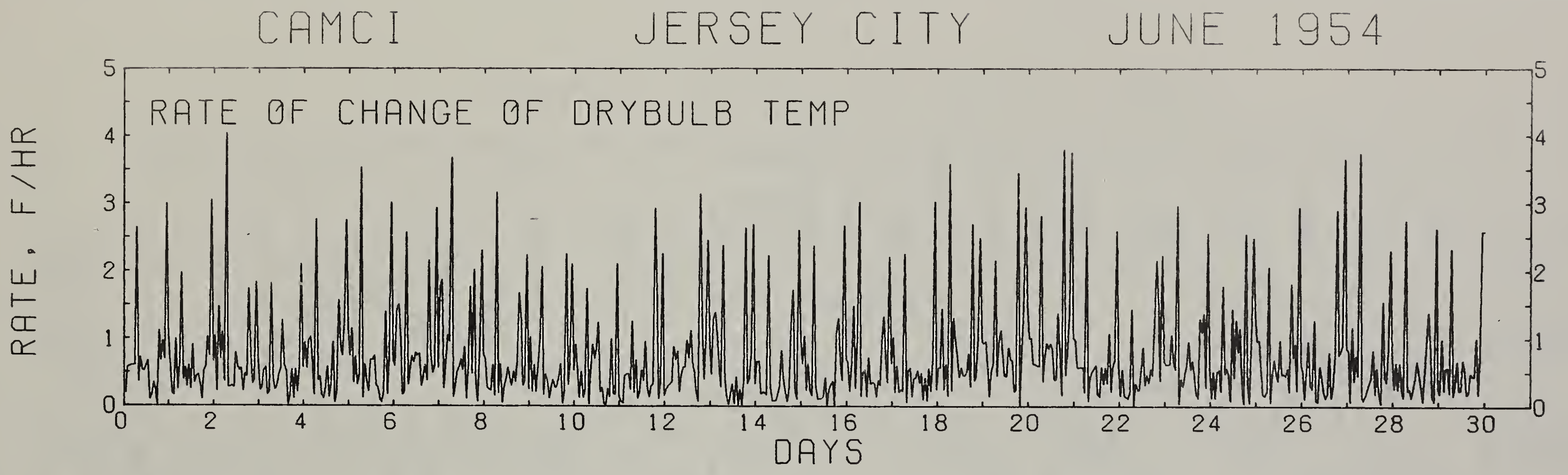

JERSEY CITY JUNE 1954
SOLAR RADIATION
BTU/HR SQFT
OUTDOOR DRYBULB AND WETBULB
TEMPERATURE
DEGREES F
DAYS
0
50
100
150
90
80
70
60
50
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Figure 32



JERSEY CITY JUNE 1954
INDOOR DRYBULB AND WETBULB
TEMPERATURE, F
110
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 33


CAMCI
JERSEY CITY
JUNE 1954
RATE OF CHANGE OF DRYBULB TEMP
RATE, F/HR
DAYS

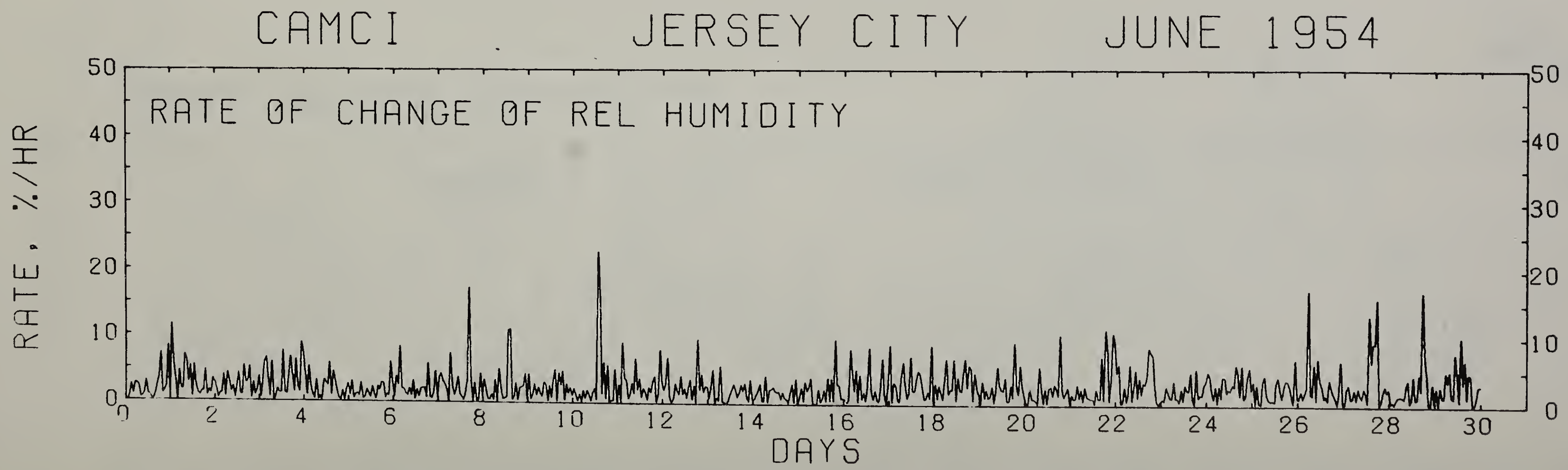

CAMCI
JERSEY CITY
JUNE 1954
RATE OF CHANGE OF REL HUMIDITY
RATE, %/HR
DAYS

Figure 34

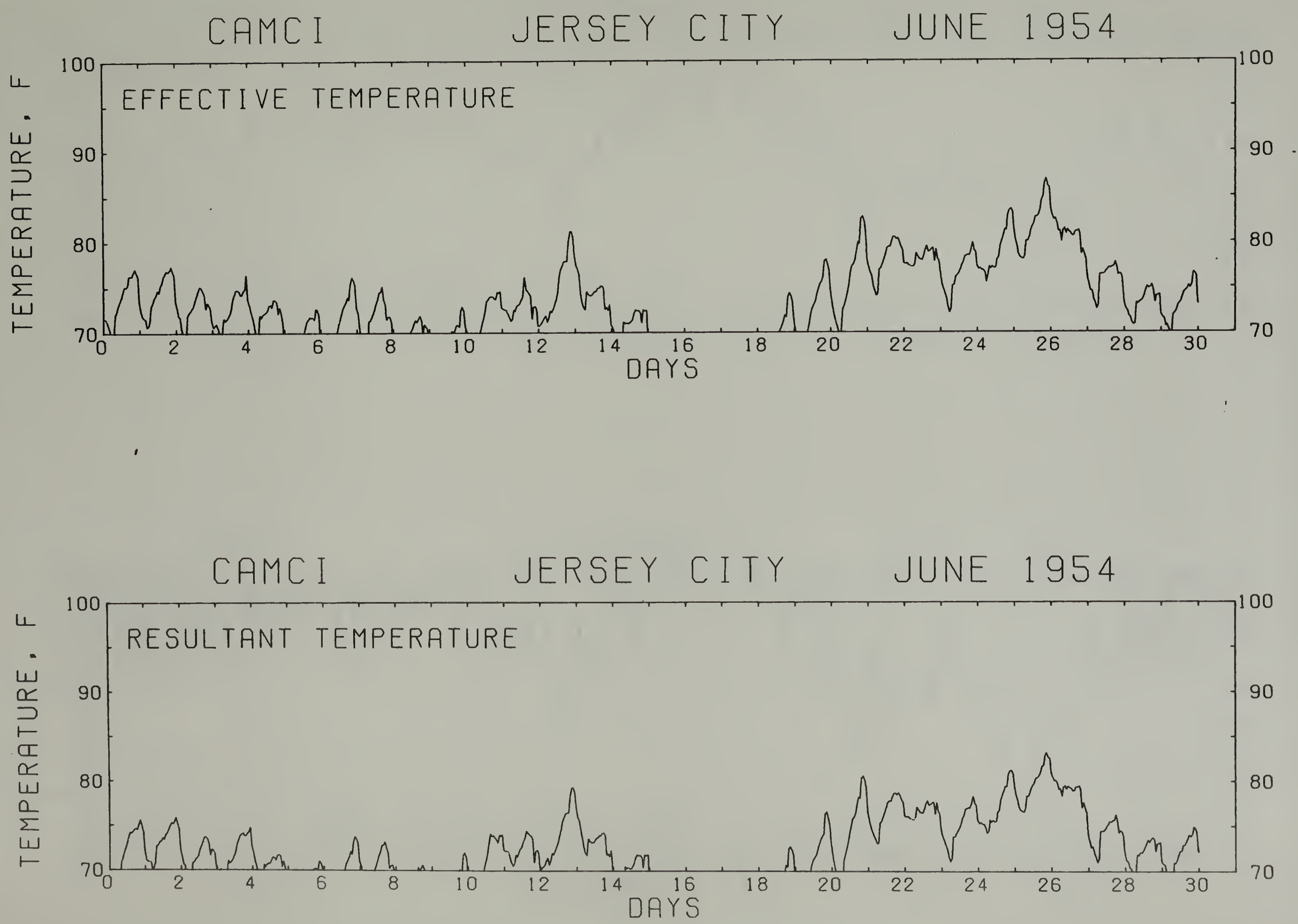

CAMCI JERSEY CITY JUNE 1954
EFFECTIVE TEMPERATURE
TEMPERATURE, F
100
90
80
70
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS
CAMCI JERSEY CITY JUNE 1954
RESULTANT TEMPERATURE
TEMPERATURE, F
100
90
80
70
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS

Figure 35

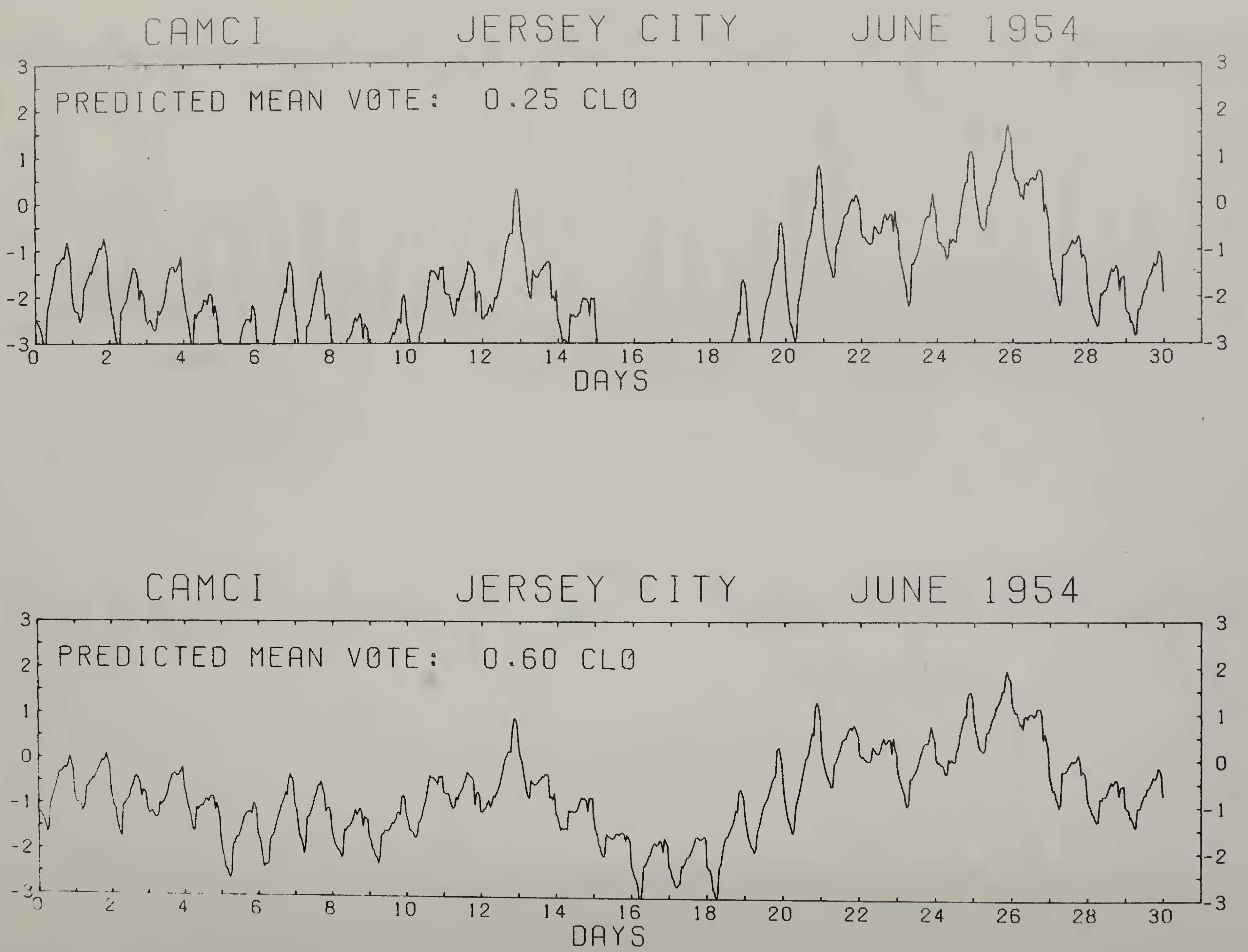

CAMCI
JERSEY CITY
JUNE 1954
PREDICTED MEAN VØTE: 0.25 CLØ
DAYS
CAMCI
JERSEY CITY
JUNE 1954
PREDICTED MEAN VØTE: 0.60 CLØ
DAYS

Figure 36

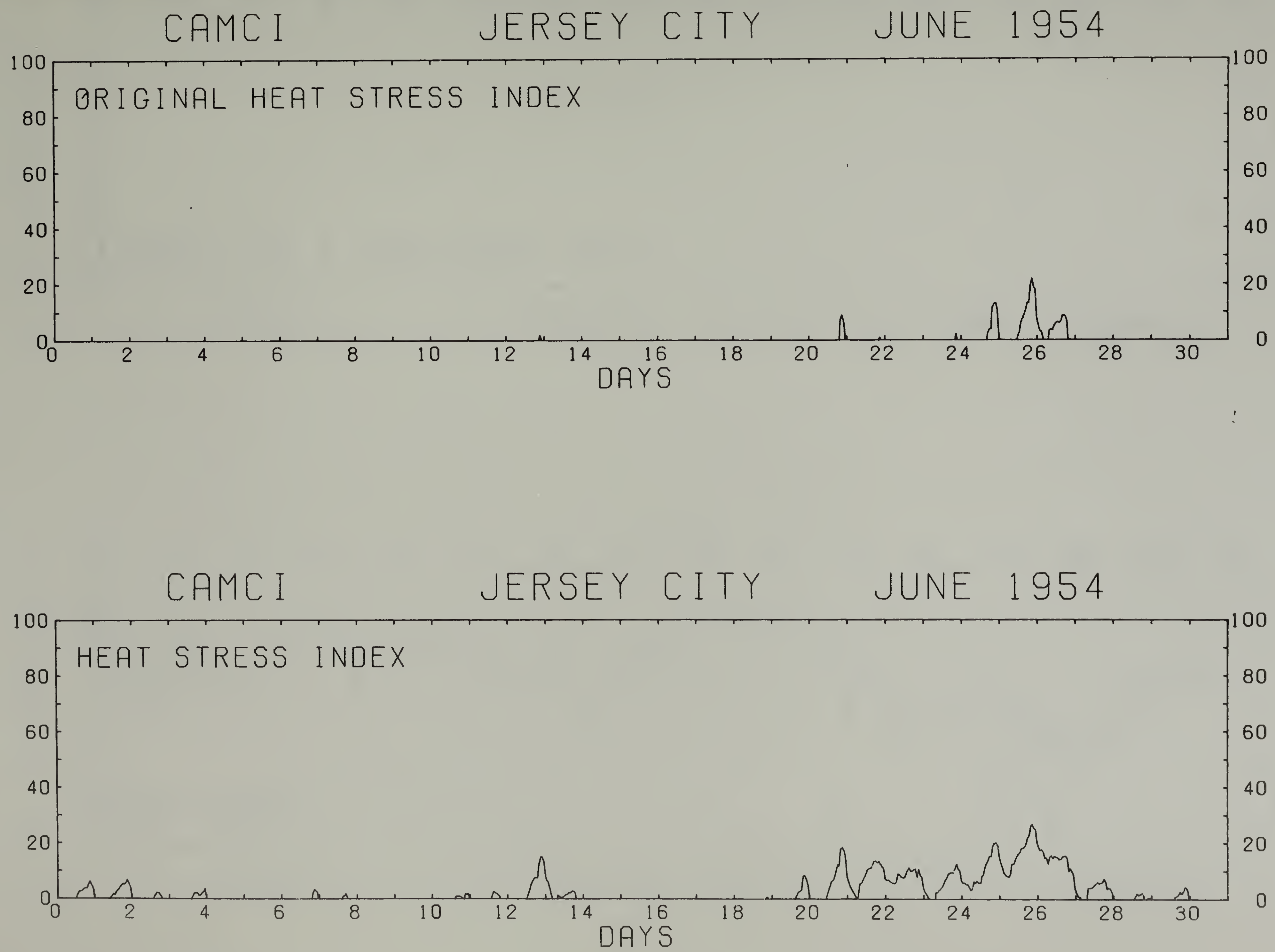

CAMCI
JERSEY CITY
JUNE 1954
ØRIGINAL HEAT STRESS INDEX
DAYS
CAMCI
JERSEY CITY
JUNE 1954
HEAT STRESS INDEX
DAYS

Figure 37

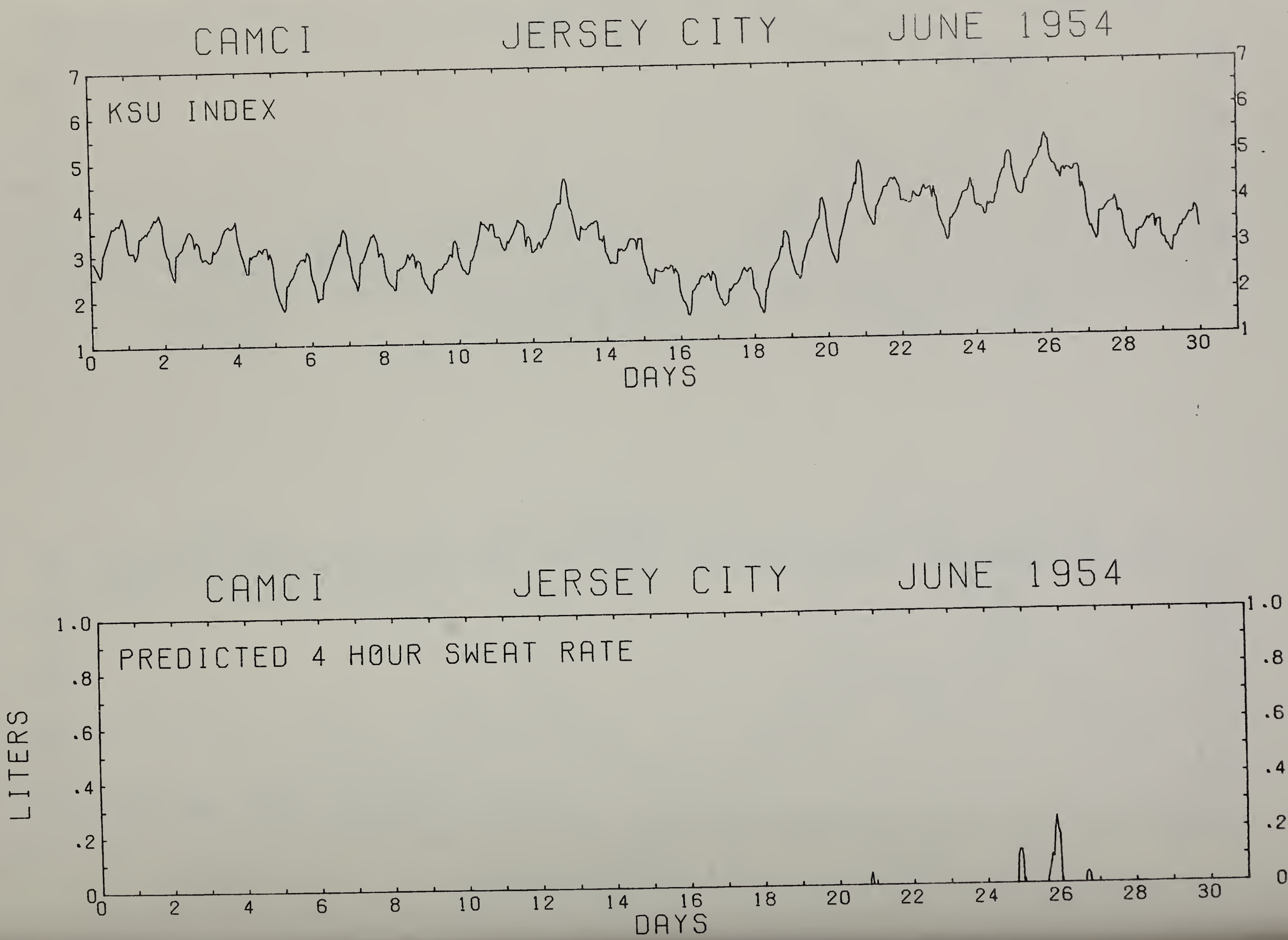

CAMCI
JERSEY CITY
JUNE 1954
KSU INDEX
DAYS
CAMCI
JERSEY CITY
JUNE 1954
PREDICTED 4 HOUR SWEAT RATE
LITERS
DAYS

Figure 38

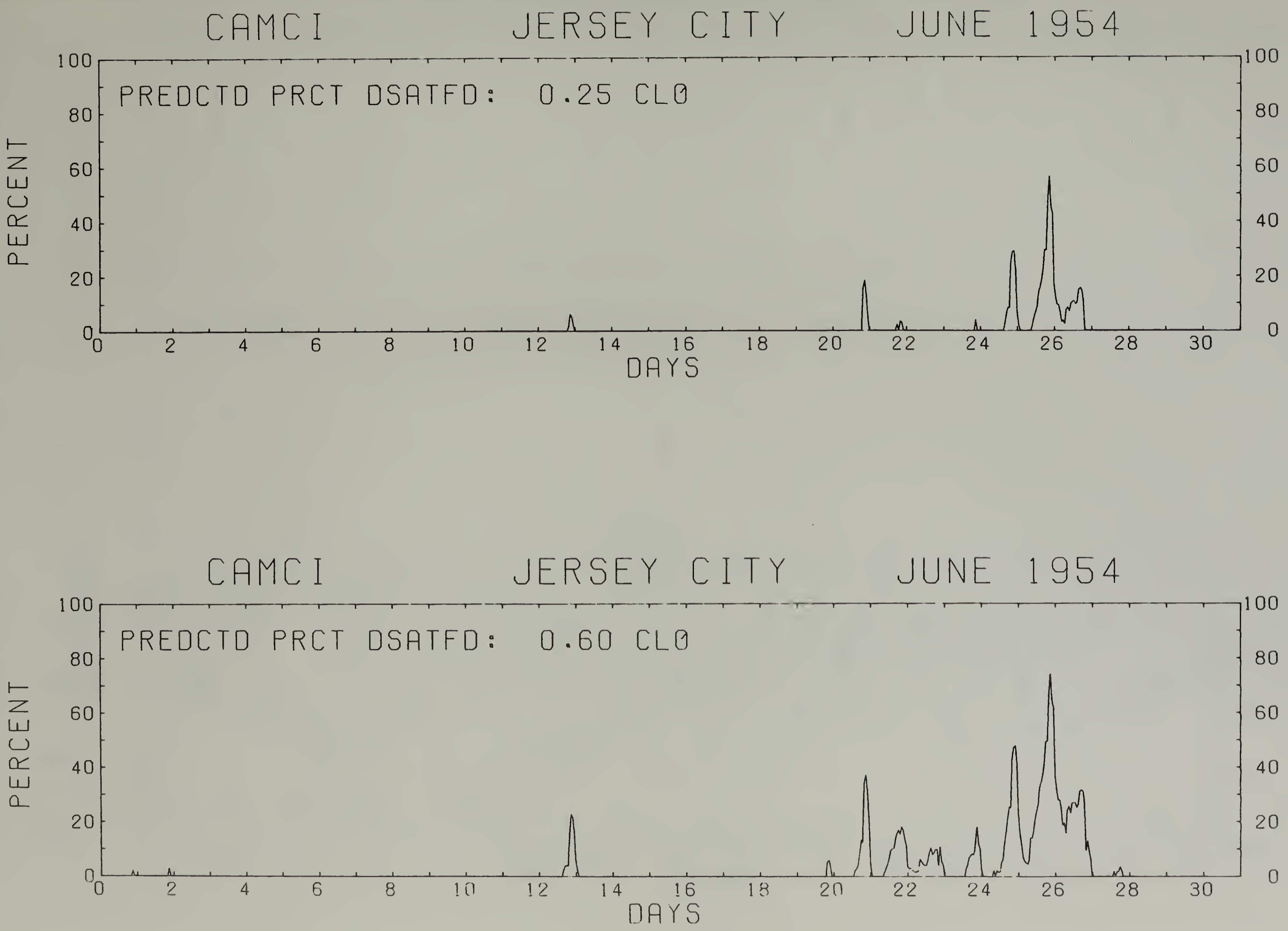

CAMCI JERSEY CITY JUNE 1954
PREDCTD PRCT DSATFD: 0.25 CLØ
PERCENT
DAYS
CAMCI JERSEY CITY JUNE 1954
PREDCTD PRCT DSATFD: 0.60 CLØ
PERCENT
DAYS

Figure 39

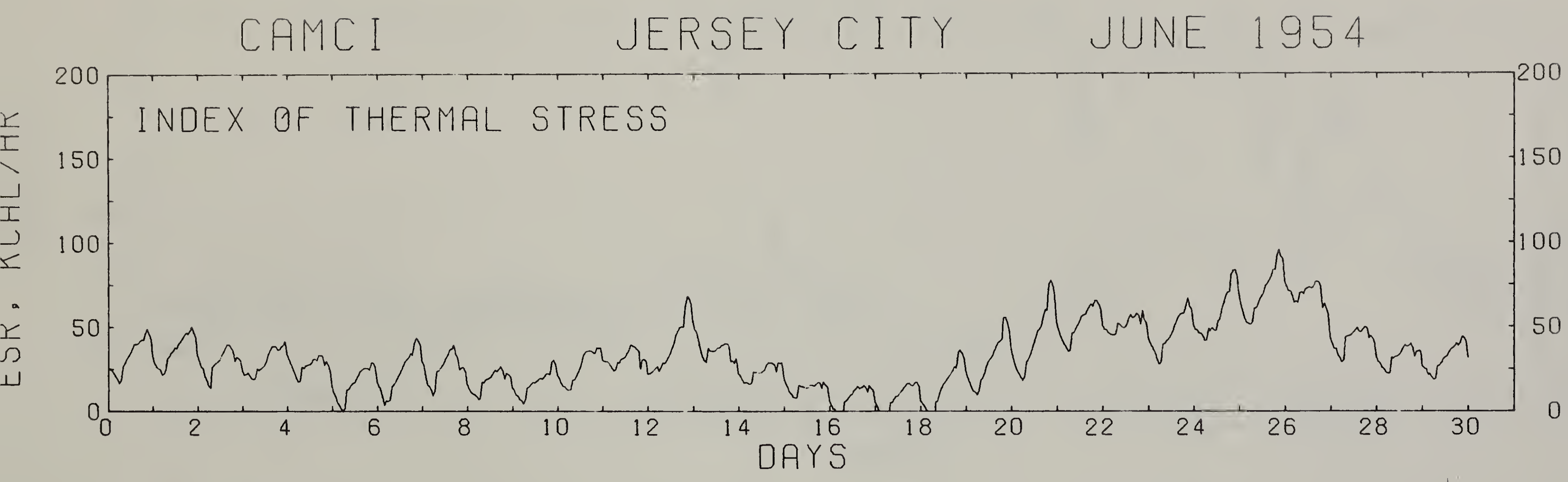

CAMCI
JERSEY CITY
JUNE 1954
INDEX OF THERMAL STRESS
ESR, KCAL/HR
DAYS
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
0
50
100
150
200

Figure 40

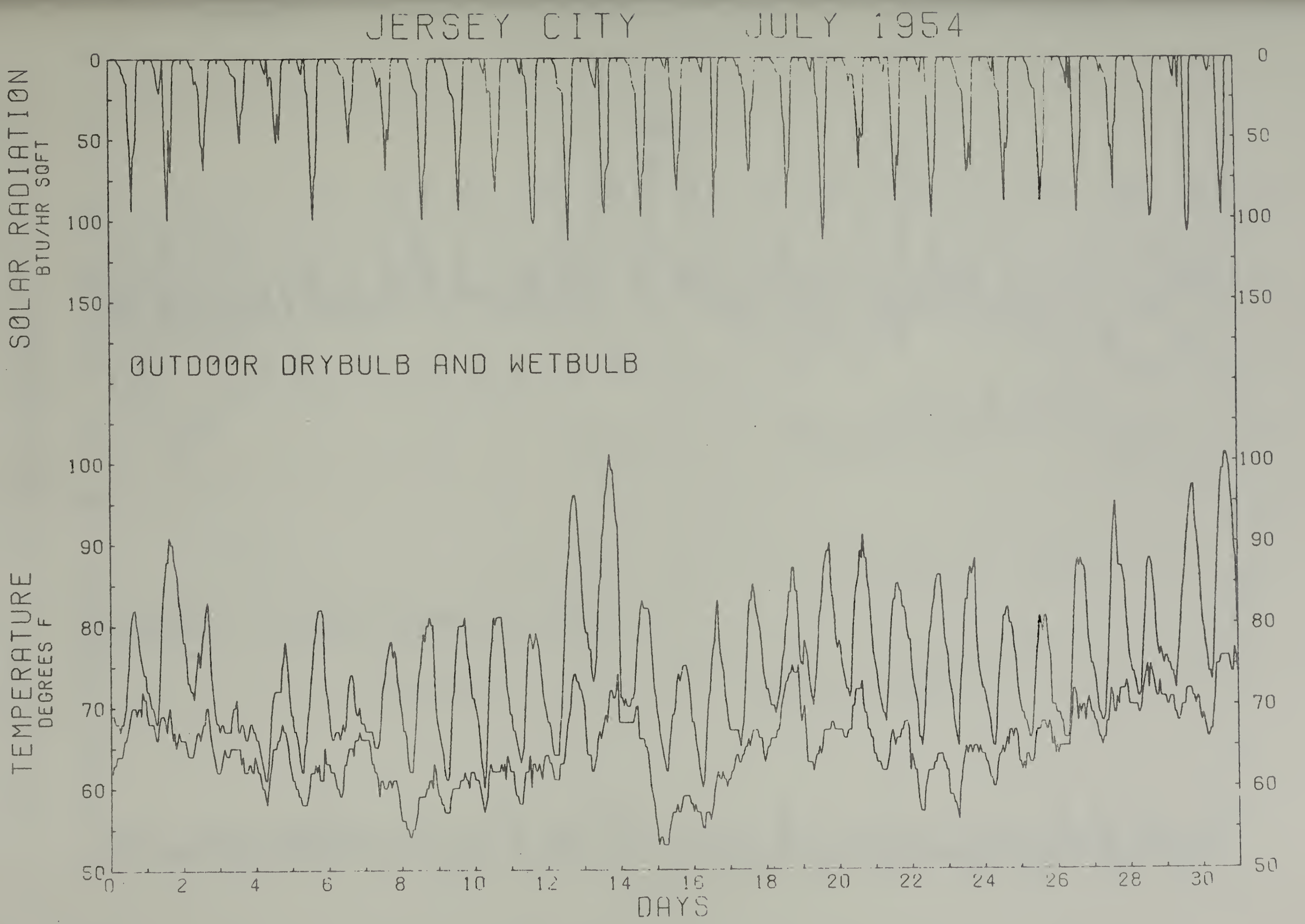

JERSEY CITY JULY 1954
SOLAR RADIATION
BTU/HR SQFT
OUTDOOR DRYBULB AND WETBULB
TEMPERATURE
DEGREES F
DAYS
0
50
100
150
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Figure 41

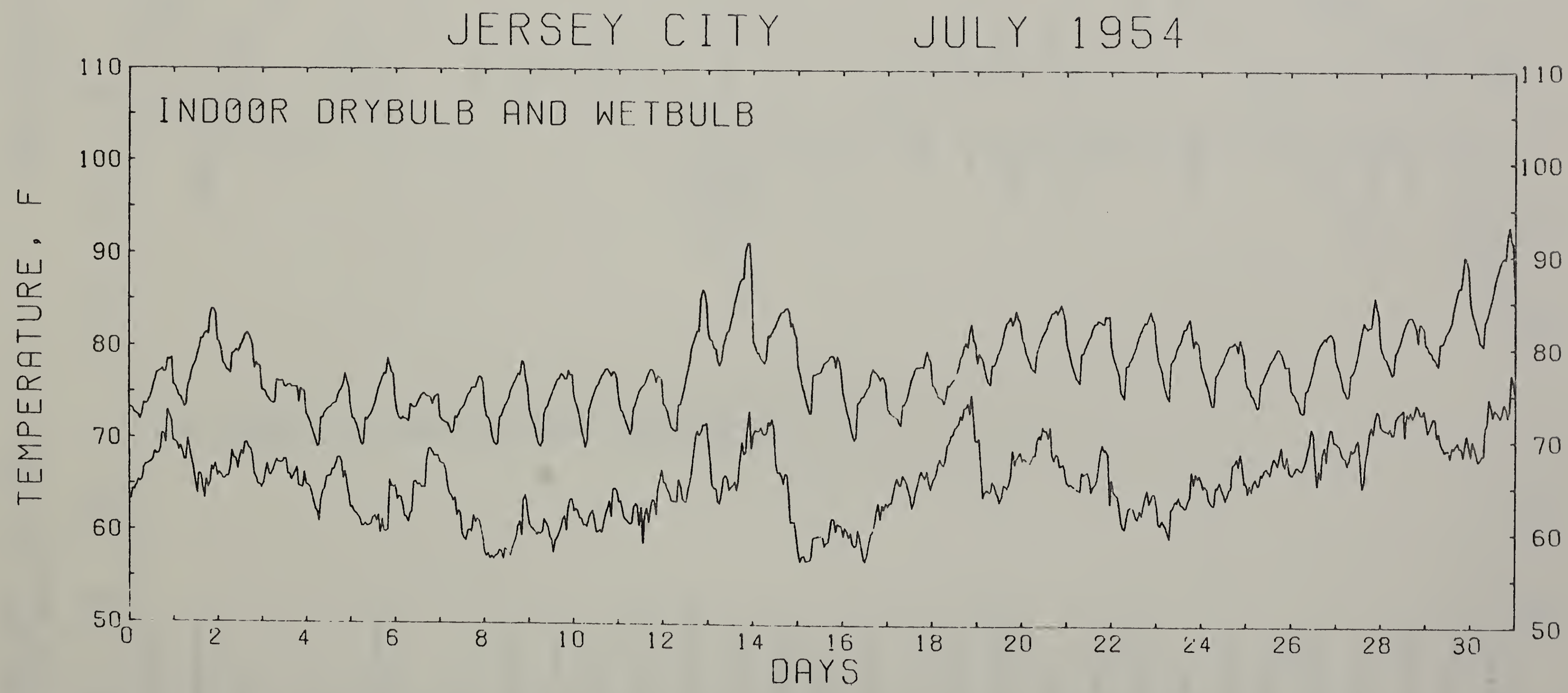

JERSEY CITY JULY 1954
INDOOR DRYBULB AND WETBULB
TEMPERATURE, F
DAYS
110
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Figure 42

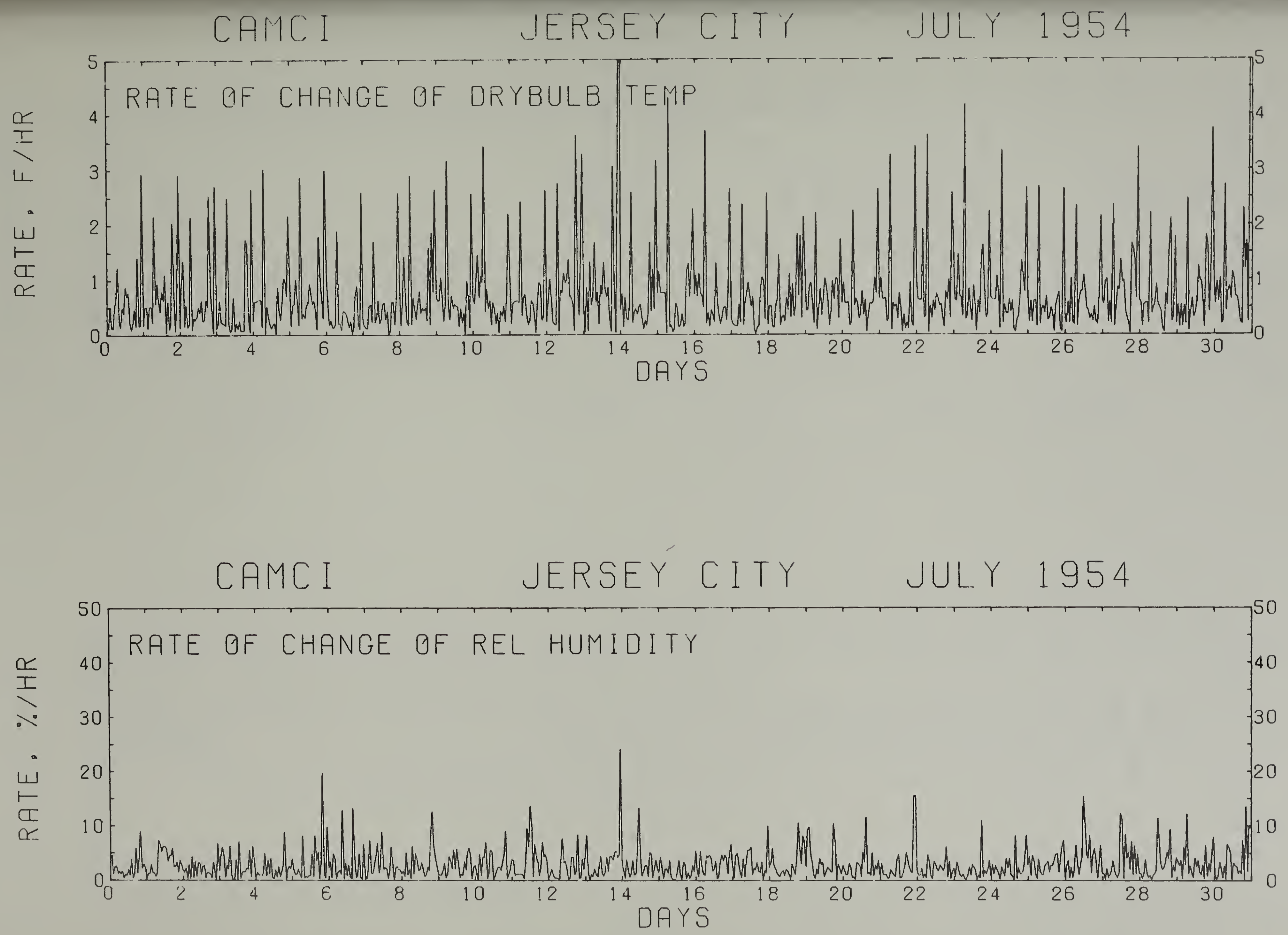
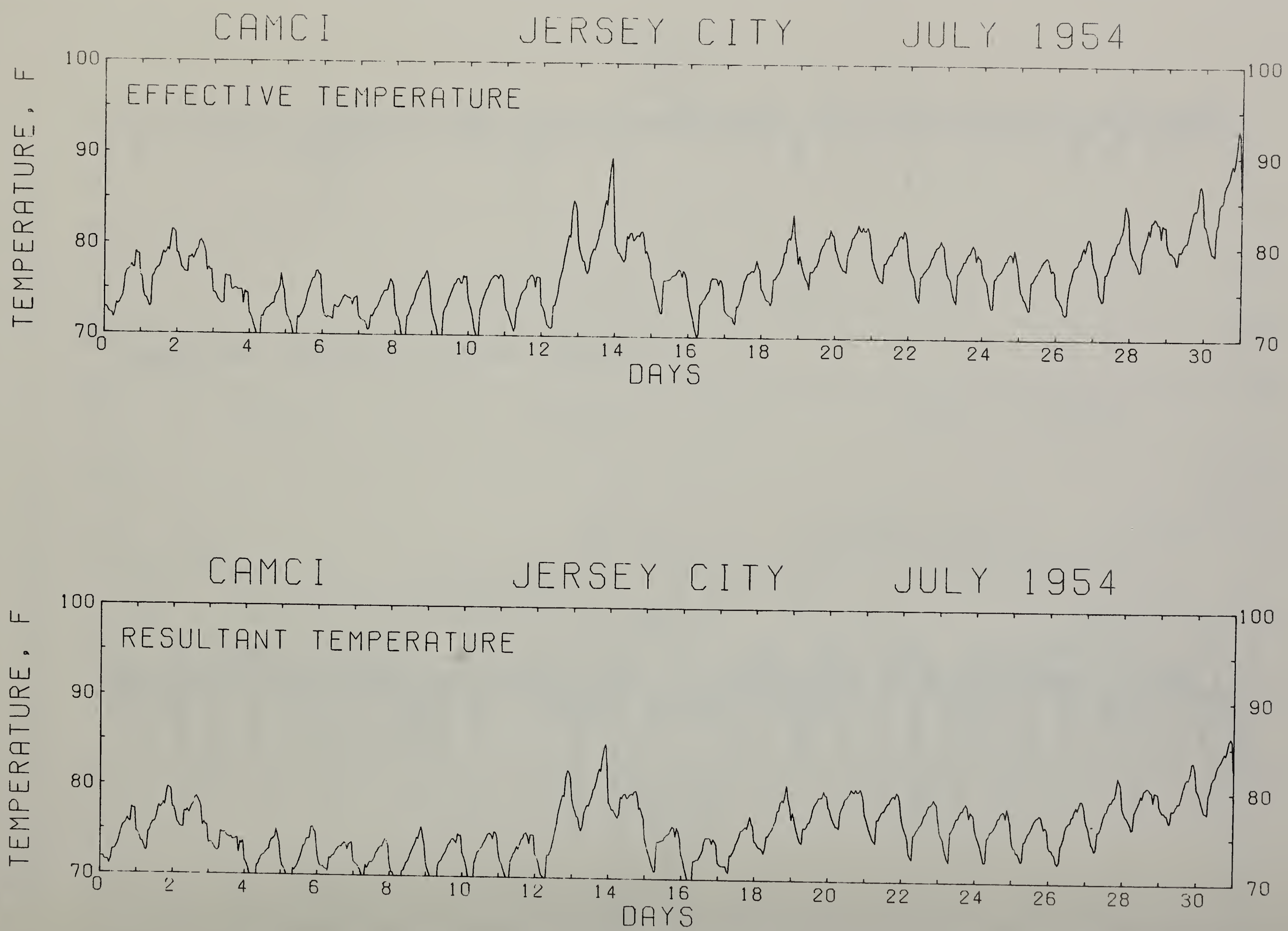

CAMCI JERSEY CITY JULY 1954
RATE OF CHANGE OF DRYBULB TEMP
RATE, F/HR
DAYS
CAMCI JERSEY CITY JULY 1954
RATE OF CHANGE OF REL HUMIDITY
RATE, %/HR
DAYS

Figure 43



CAMCI
JERSEY CITY
JULY 1954
EFFECTIVE TEMPERATURE
TEMPERATURE, F
DAYS
100
90
80
70
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
CAMCI
JERSEY CITY
JULY 1954
RESULTANT TEMPERATURE
TEMPERATURE, F
DAYS

Figure 44

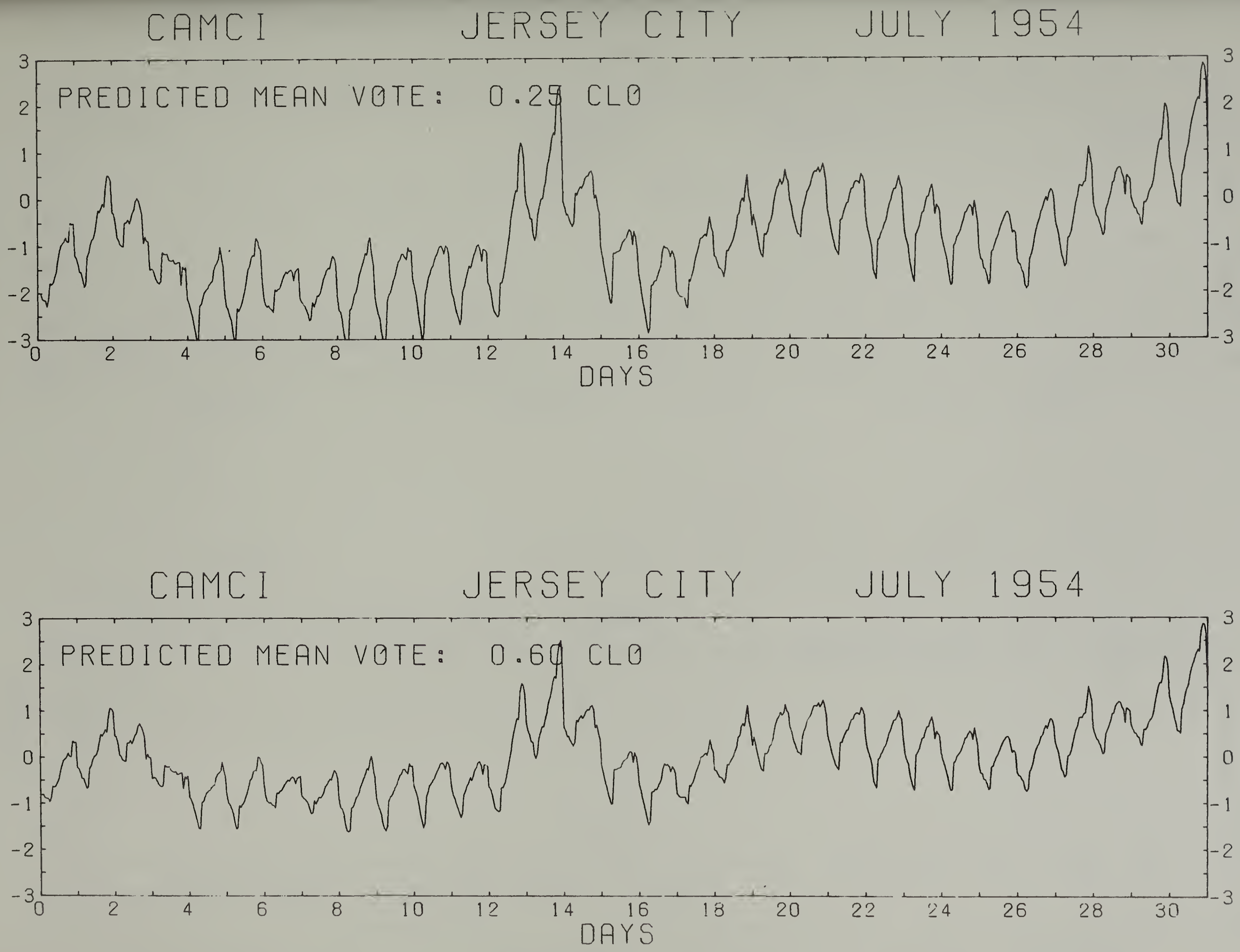
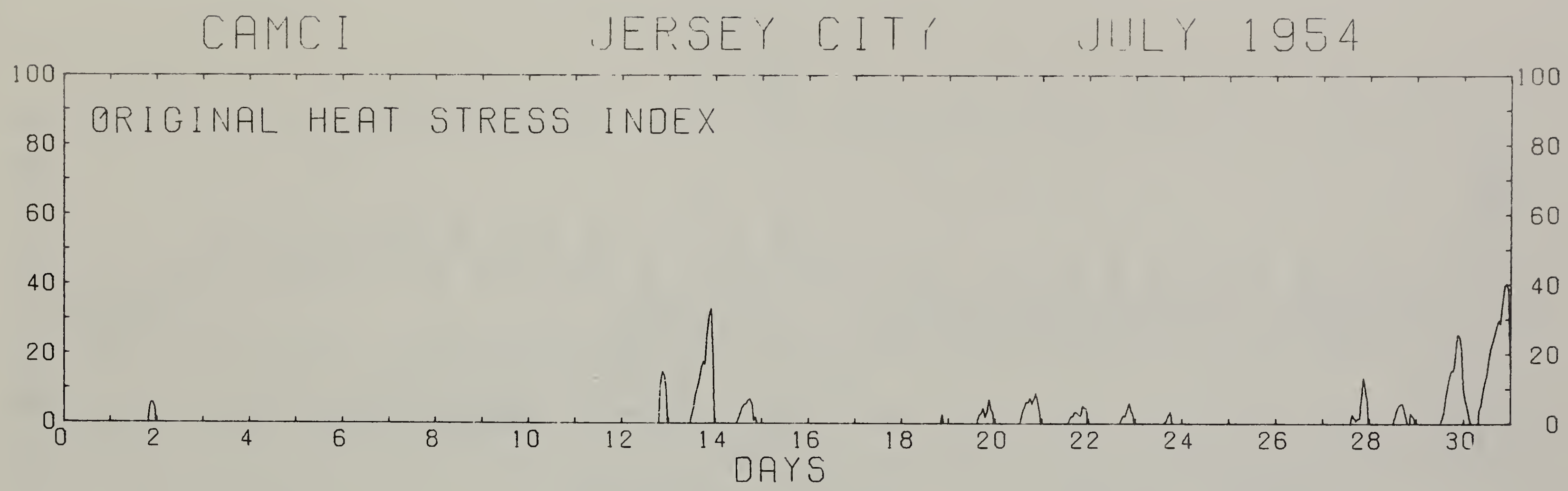

CAMCI
JERSEY CITY
JULY 1954
PREDICTED MEAN VØTE: 0.25 CLØ
DAYS
CAMCI
JERSEY CITY
JULY 1954
PREDICTED MEAN VØTE: 0.60 CLØ
DAYS

Figure 45


CAMCI
JERSEY CITY
JULY 1954
ØRIGINAL HEAT STRESS INDEX
DAYS
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
100
80
60
40
20
0

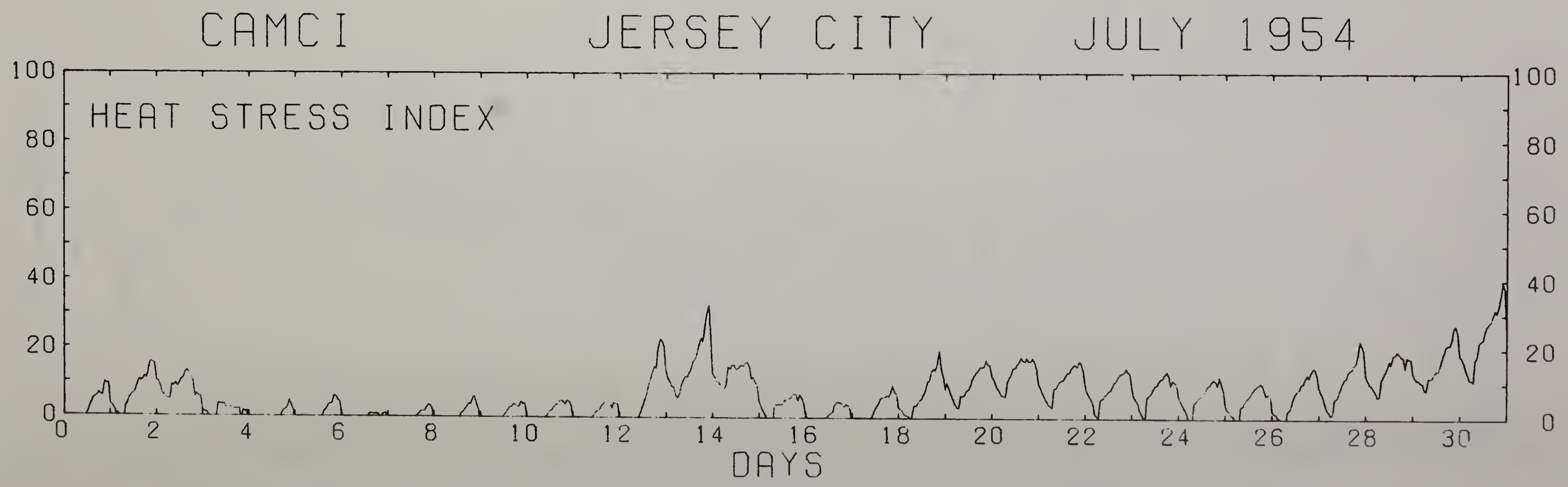
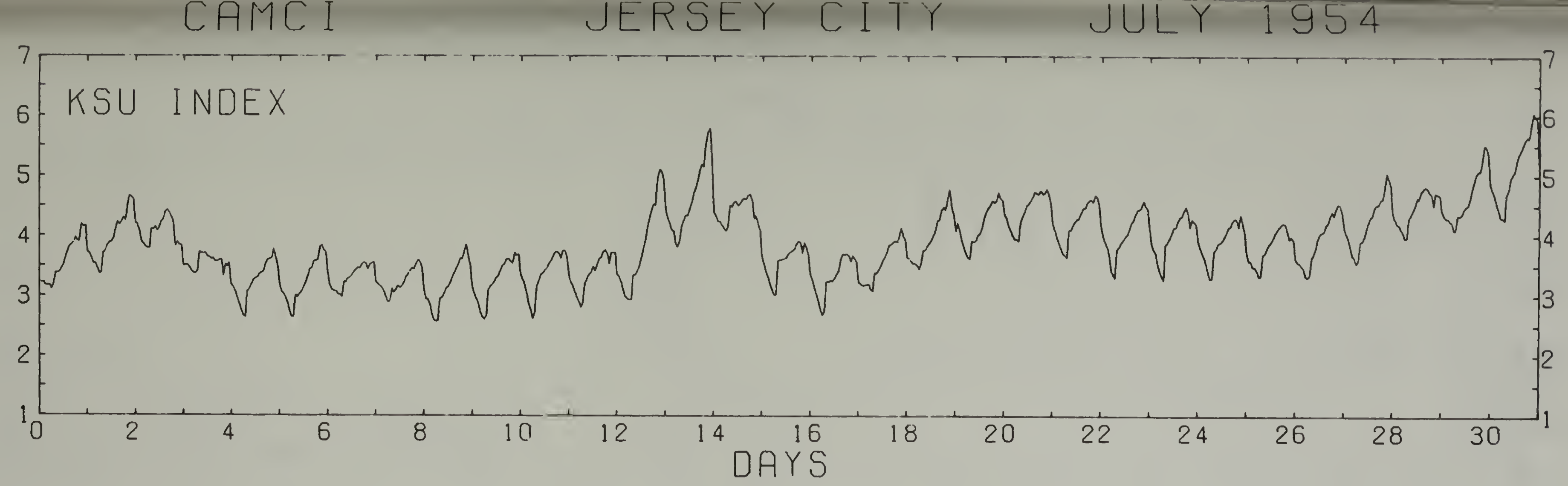

CAMCI
JERSEY CITY
JULY 1954
HEAT STRESS INDEX
DAYS
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
100
80
60
40
20
0

Figure 46


CAMCI
JERSEY CITY
JULY 1954
KSU INDEX
DAYS

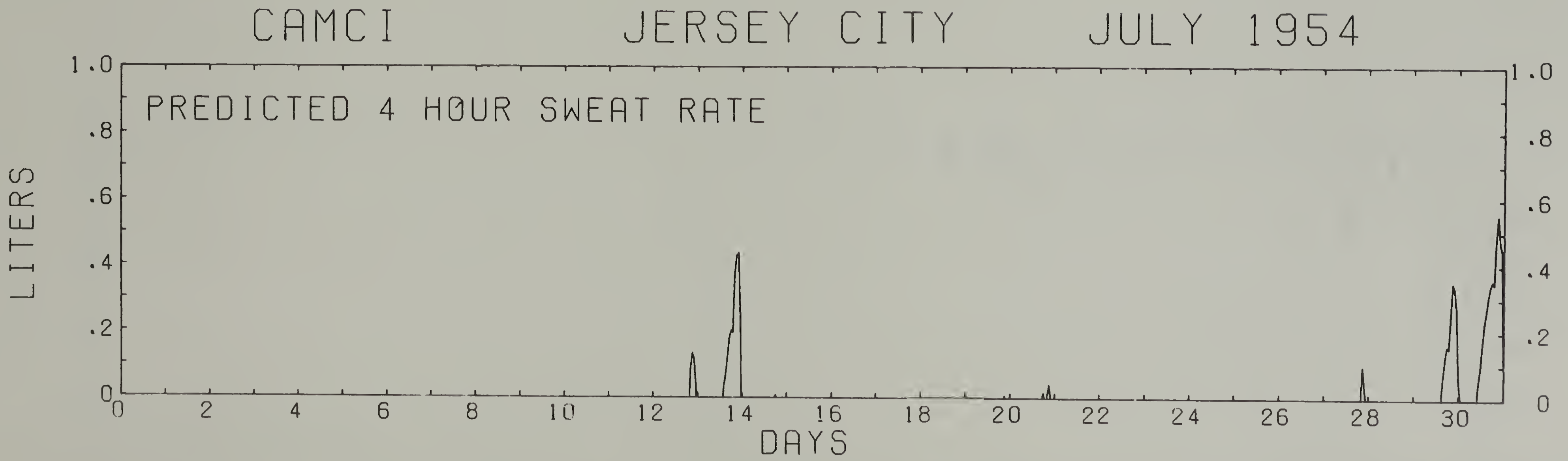

CAMCI
JERSEY CITY
JULY 1954
PREDICTED 4 HØUR SWEAT RATE
LITERS
DAYS

Figure 47

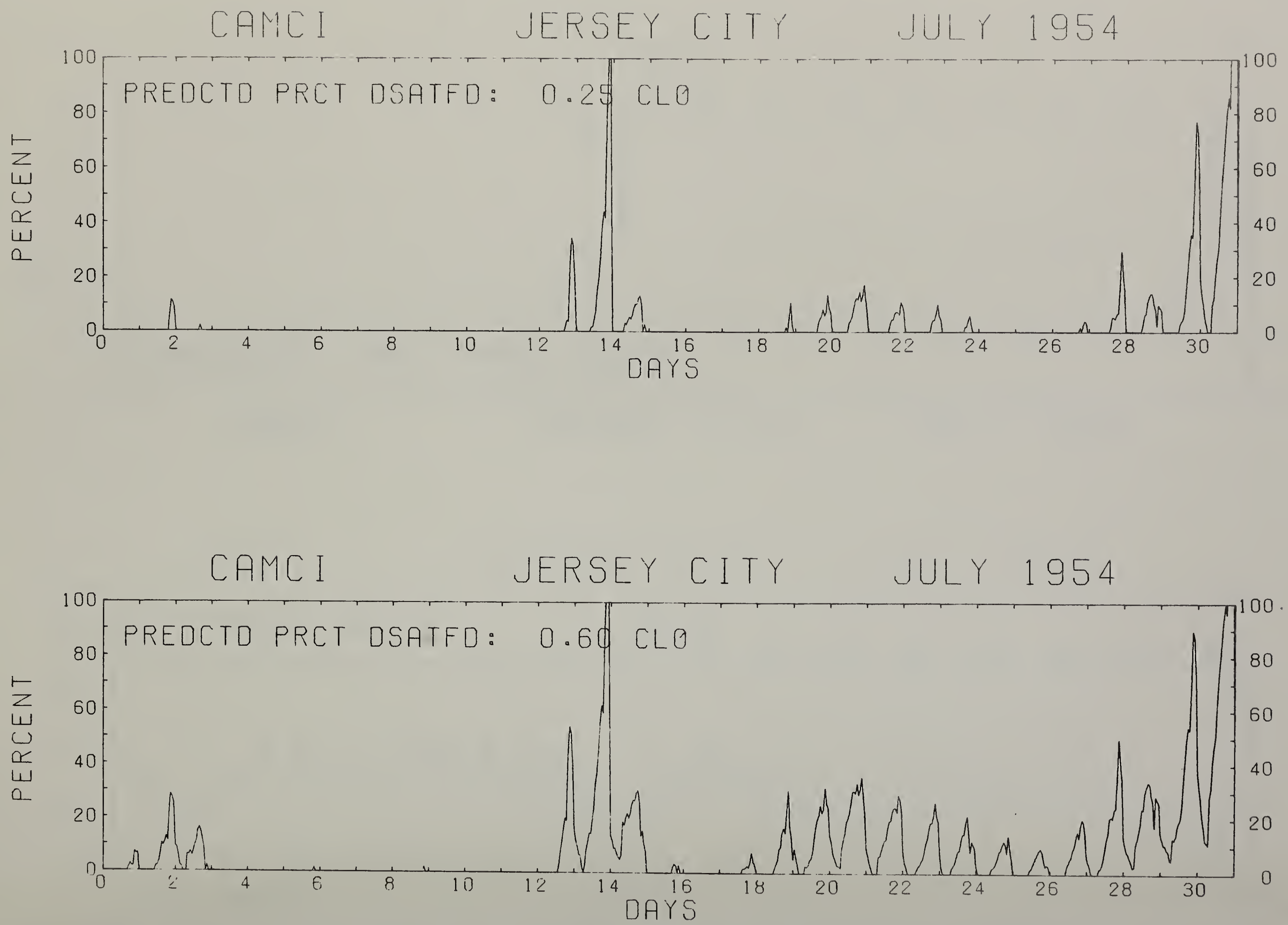

CAMCI
JERSEY CITY
JULY 1954
PREDCTD PRCT DSATFD: 0.25 CLO
PERCENT
DAYS
CAMCI
JERSEY CITY
JULY 1954
PREDCTD PRCT DSATFD: 0.60 CLO
PERCENT
DAYS

Figure 48

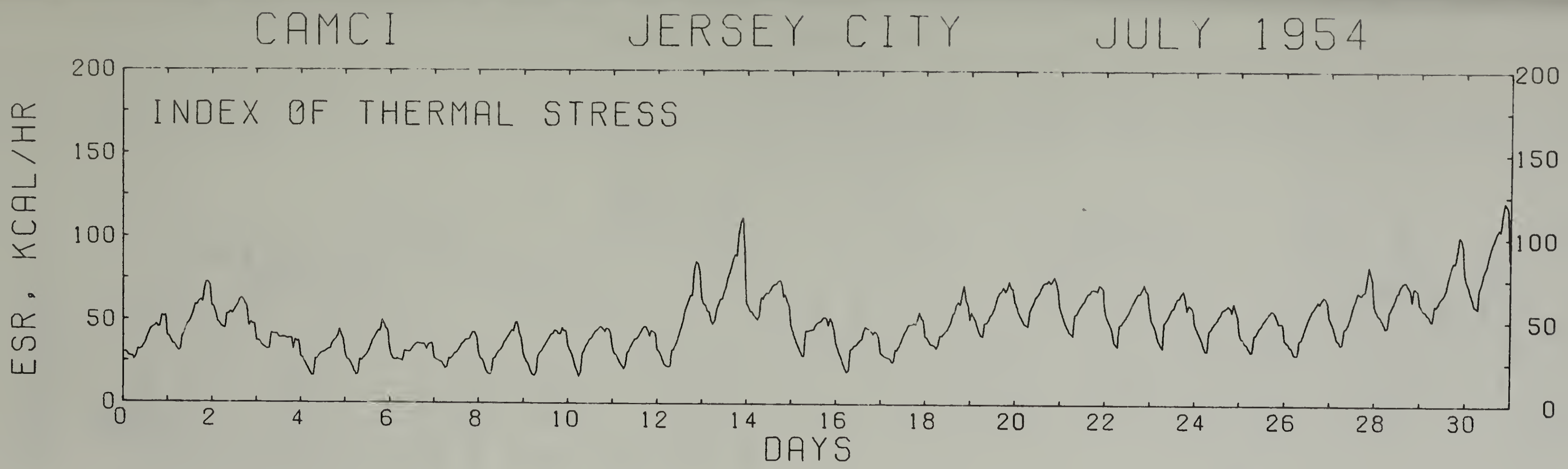

CAMCI
JERSEY CITY
JULY 1954
INDEX ØF THERMAL STRESS
ESR, KCAL/HR
DAYS
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
50
100
150
200

Figure 49

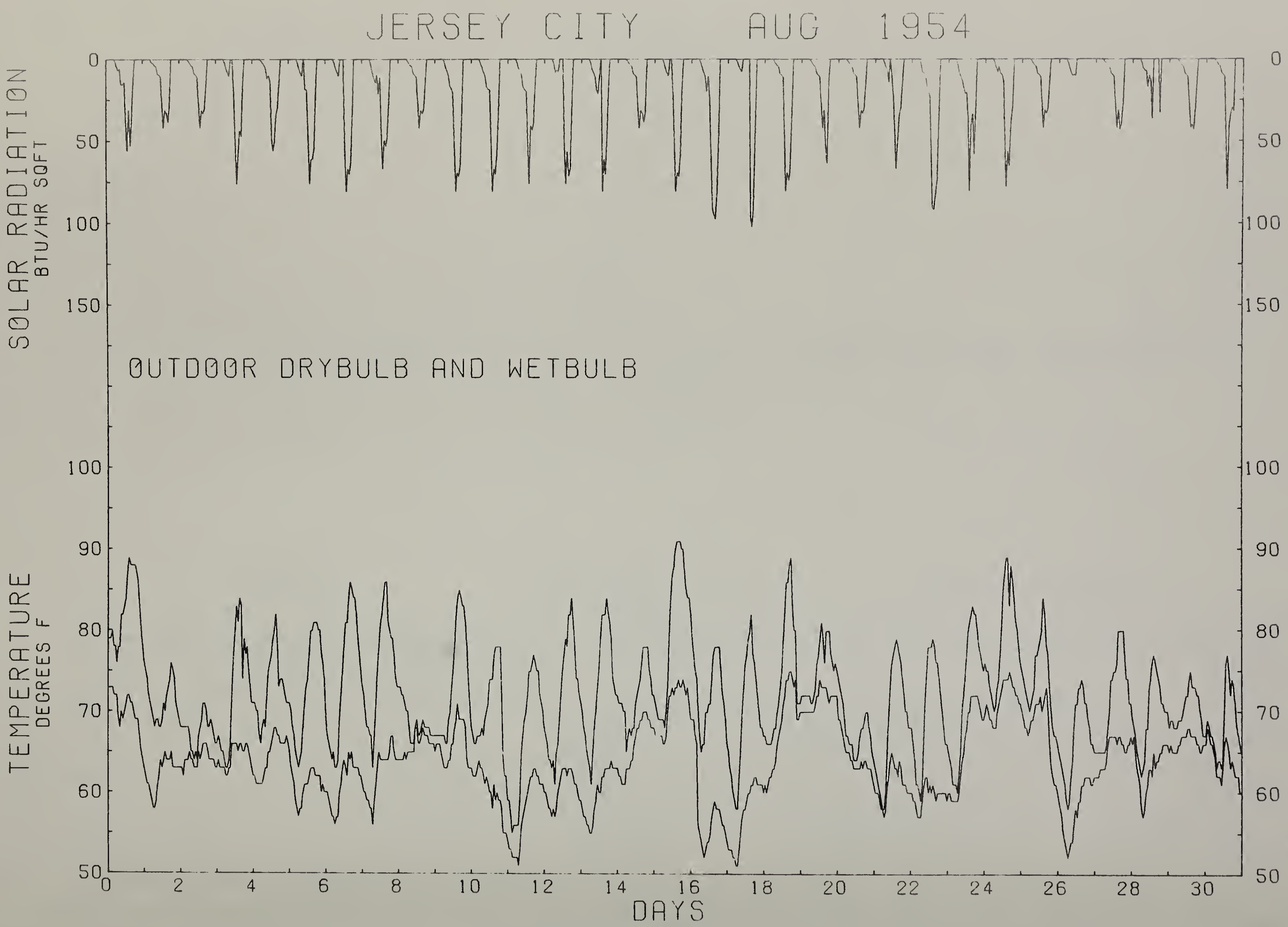

JERSEY CITY AUG 1954
SØLAR RADIATIØN
BTU/HR SQFT
ØUTDØØR DRYBULB AND WETBULB
TEMPERATURE
DEGREES F
DAYS
0
50
100
150
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Figure 50

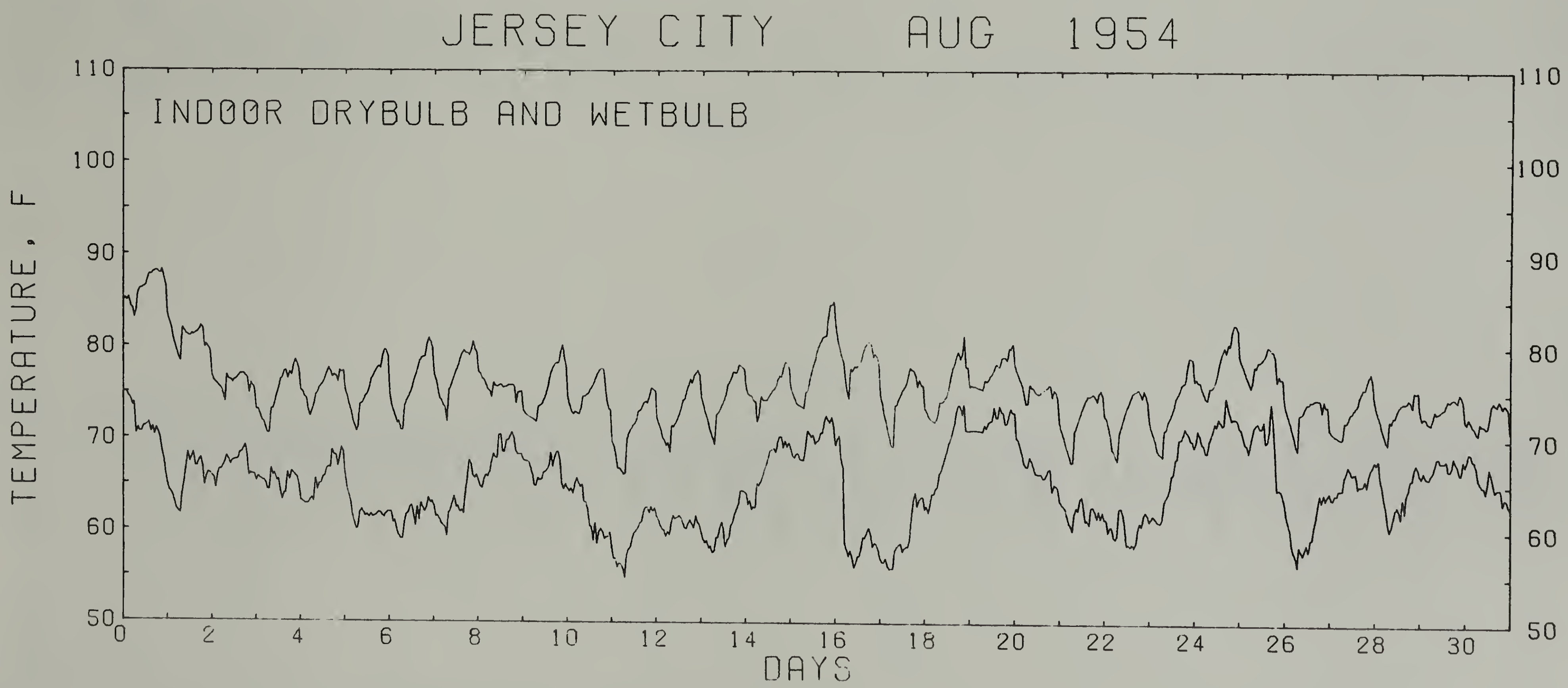
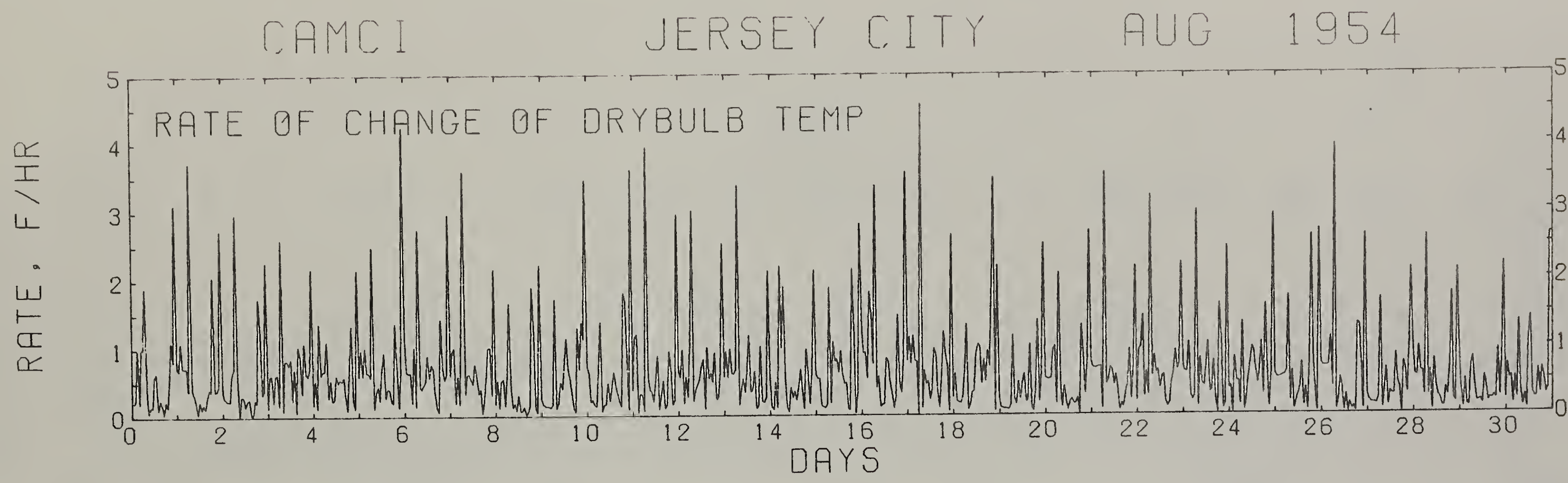

JERSEY CITY AUG 1954
INDOOR DRYBULB AND WETBULB
TEMPERATURE, F
110
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 51


CAMCI JERSEY CITY AUG 1954
RATE OF CHANGE OF DRYBULB TEMP
RATE, F/HR
0
1
2
3
4
5
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

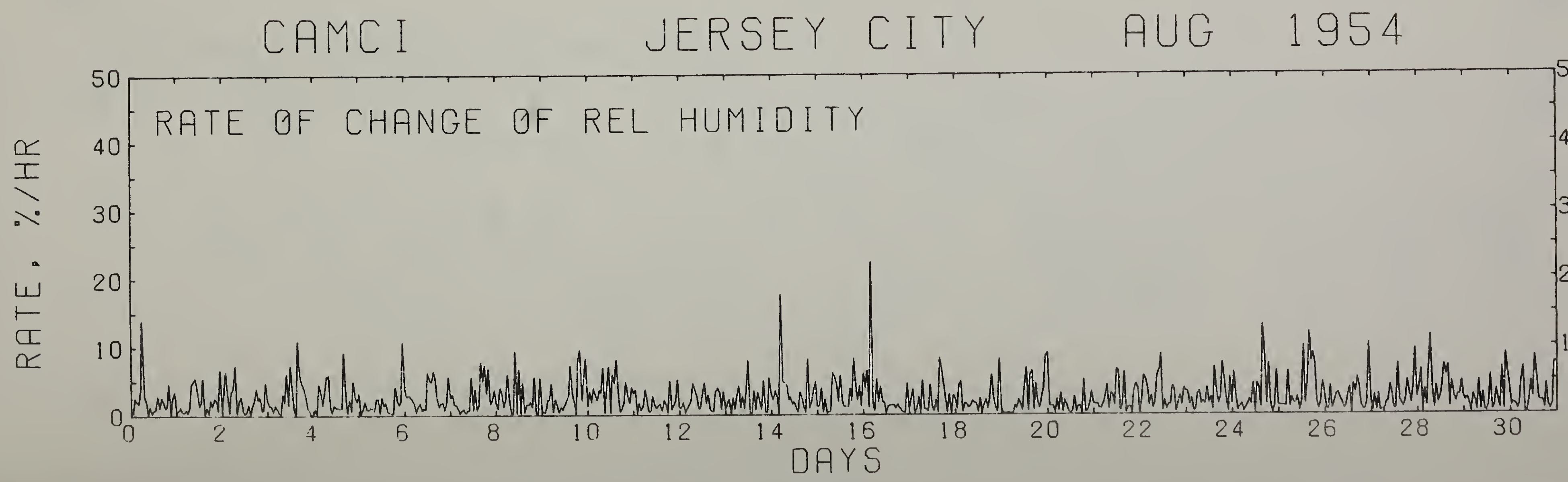

CAMCI JERSEY CITY AUG 1954
RATE OF CHANGE OF REL HUMIDITY
RATE, %/HR
0
10
20
30
40
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 52

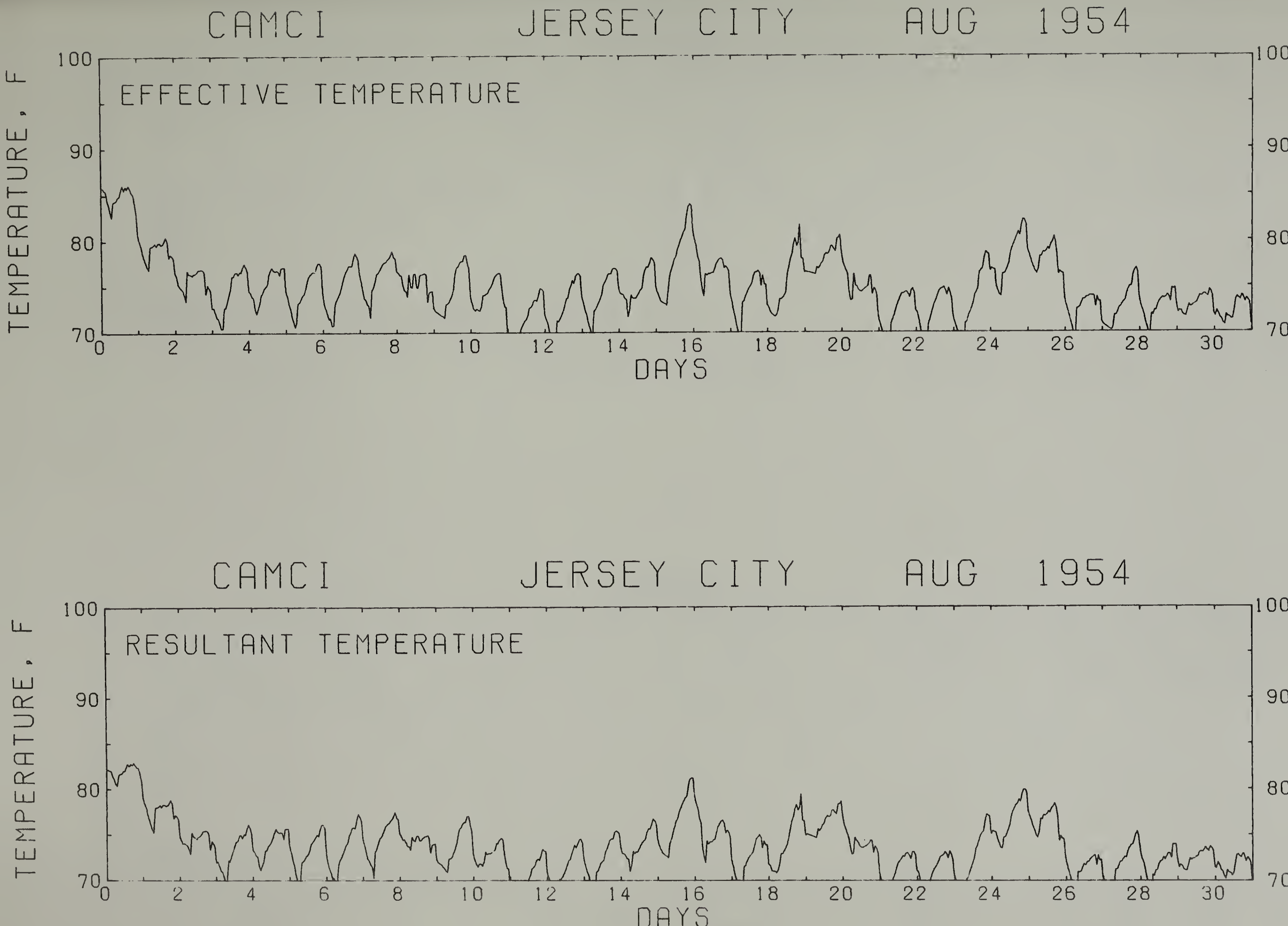
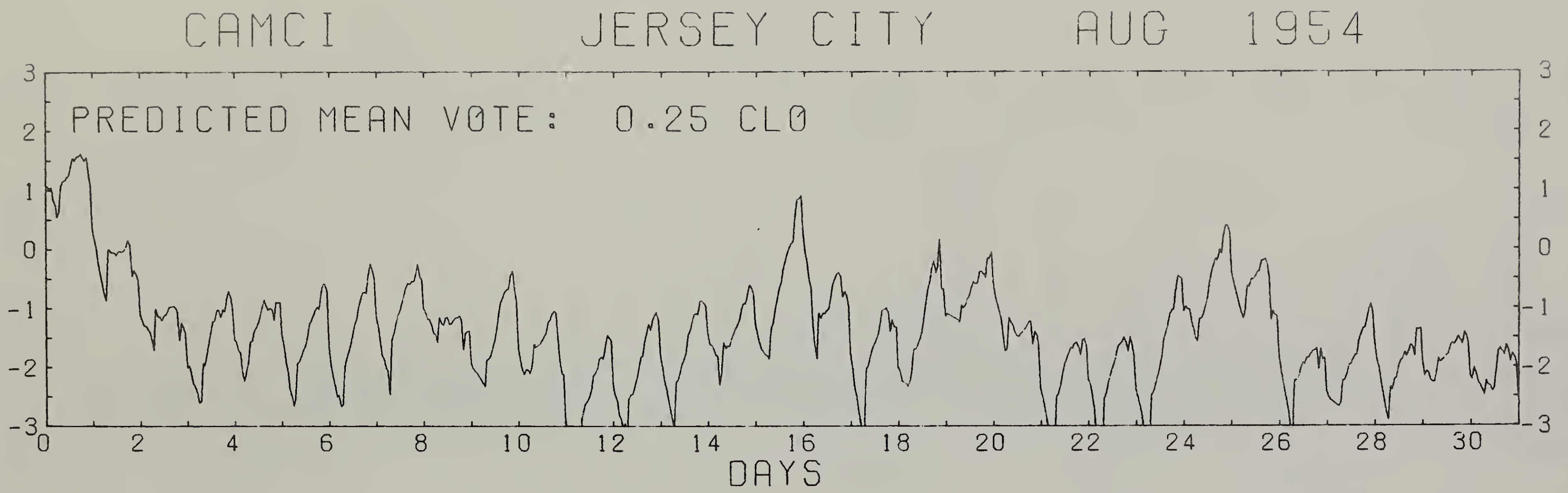

CAMCI JERSEY CITY AUG 1954
EFFECTIVE TEMPERATURE
TEMPERATURE, F
100
90
80
70
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS
CAMCI JERSEY CITY AUG 1954
RESULTANT TEMPERATURE
TEMPERATURE, F
100
90
80
70
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS

Figure 53


CAMCI JERSEY CITY AUG 1954
PREDICTED MEAN VØTE: 0.25 CLØ
3
2
1
0
-1
-2
-3
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

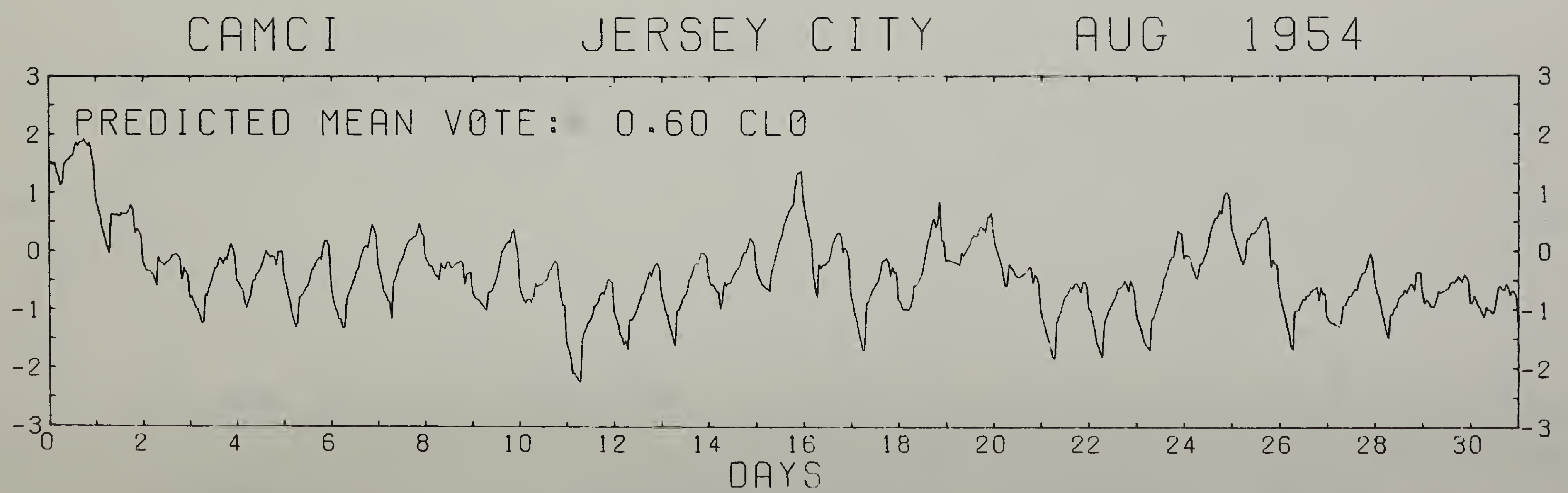
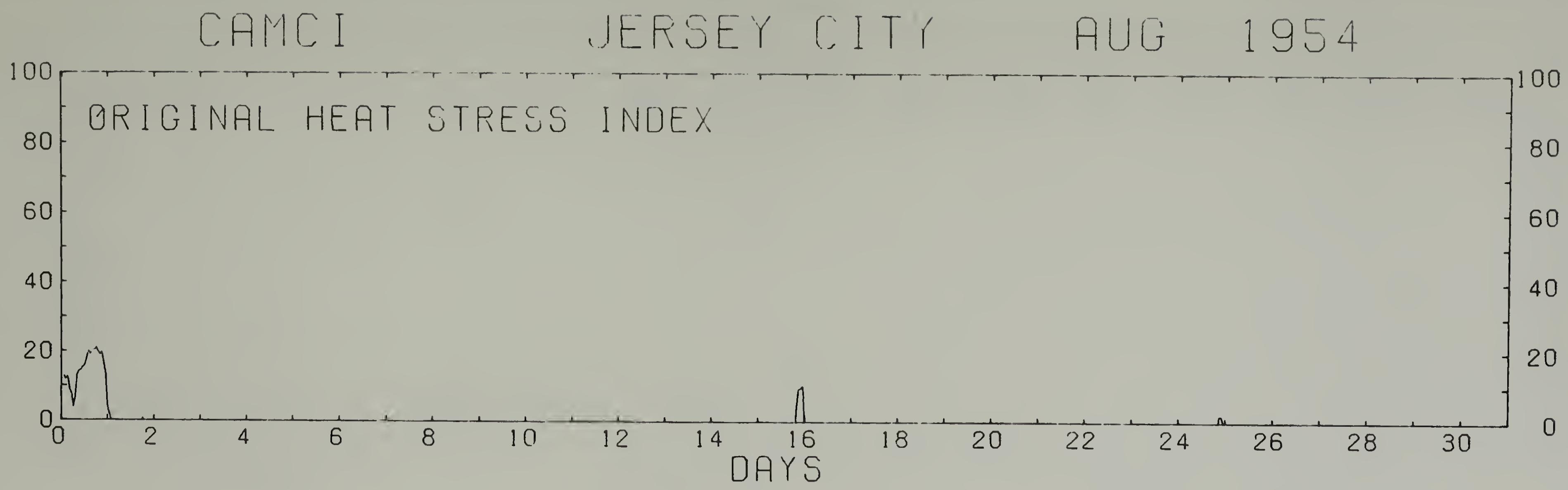

CAMCI JERSEY CITY AUG 1954
PREDICTED MEAN VØTE: 0.60 CLØ
3
2
1
0
-1
-2
-3
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 54


CAMCI
JERSEY CITY
AUG 1954
ORIGINAL HEAT STRESS INDEX
100
80
60
40
20
0
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS

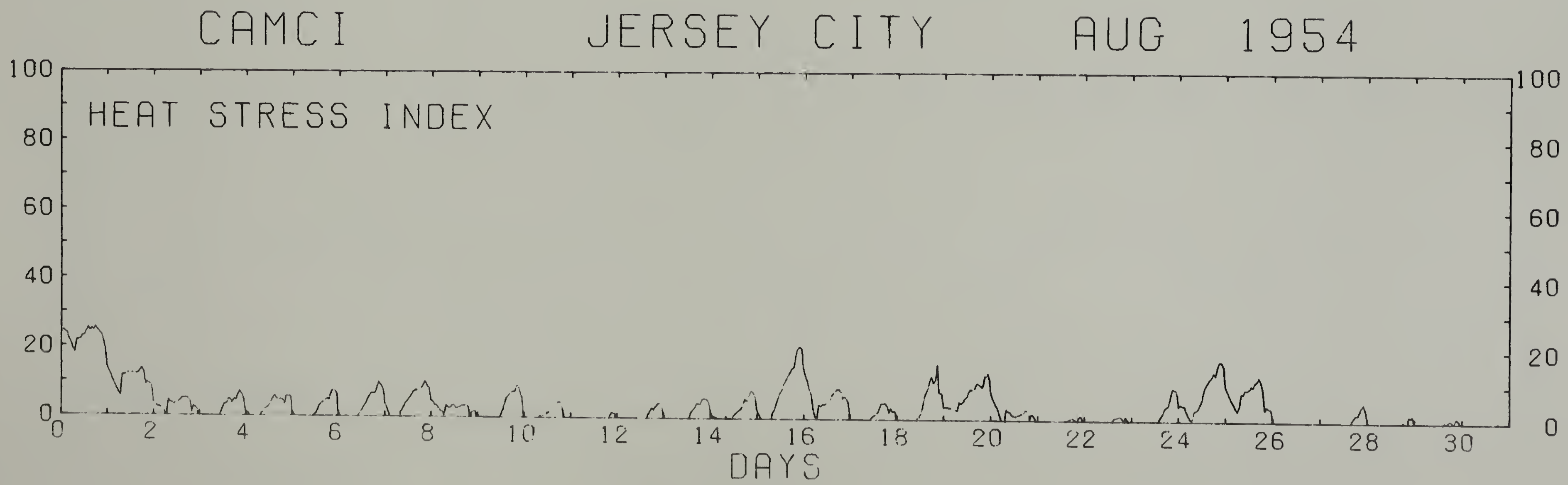

CAMCI
JERSEY CITY
AUG 1954
HEAT STRESS INDEX
100
80
60
40
20
0
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
DAYS

Figure 55

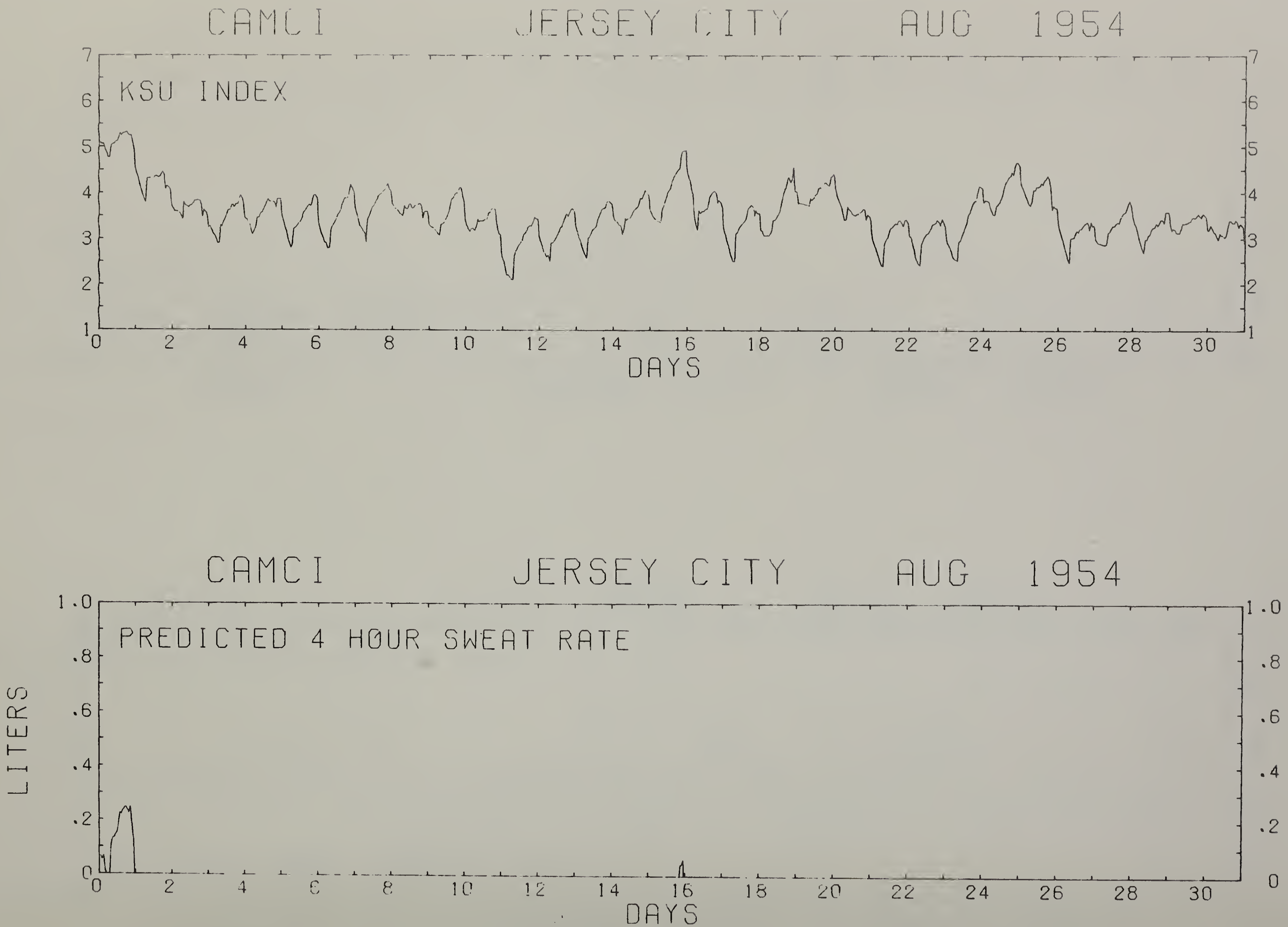

CAMCI JERSEY CITY AUG 1954
KSU INDEX
DAYS
CAMCI JERSEY CITY AUG 1954
PREDICTED 4 HOUR SWEAT RATE
LITERS
DAYS

Figure 56

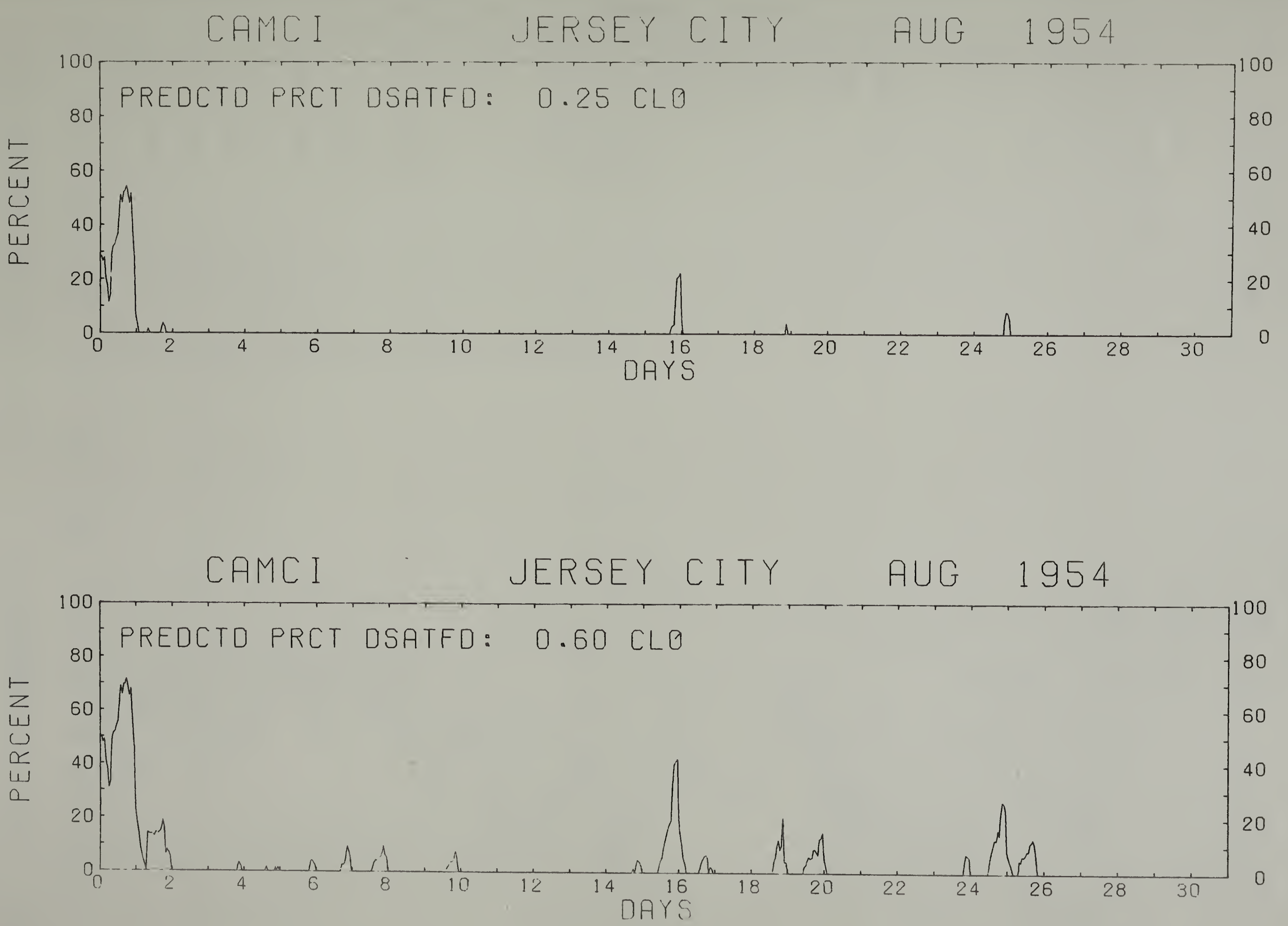

CAMCI JERSEY CITY AUG 1954
PREDCTD PRCT DSATFD: 0.25 CLØ
PERCENT
DAYS
CAMCI JERSEY CITY AUG 1954
PREDCTD PRCT DSATFD: 0.60 CLØ
PERCENT
DAYS

Figure 57

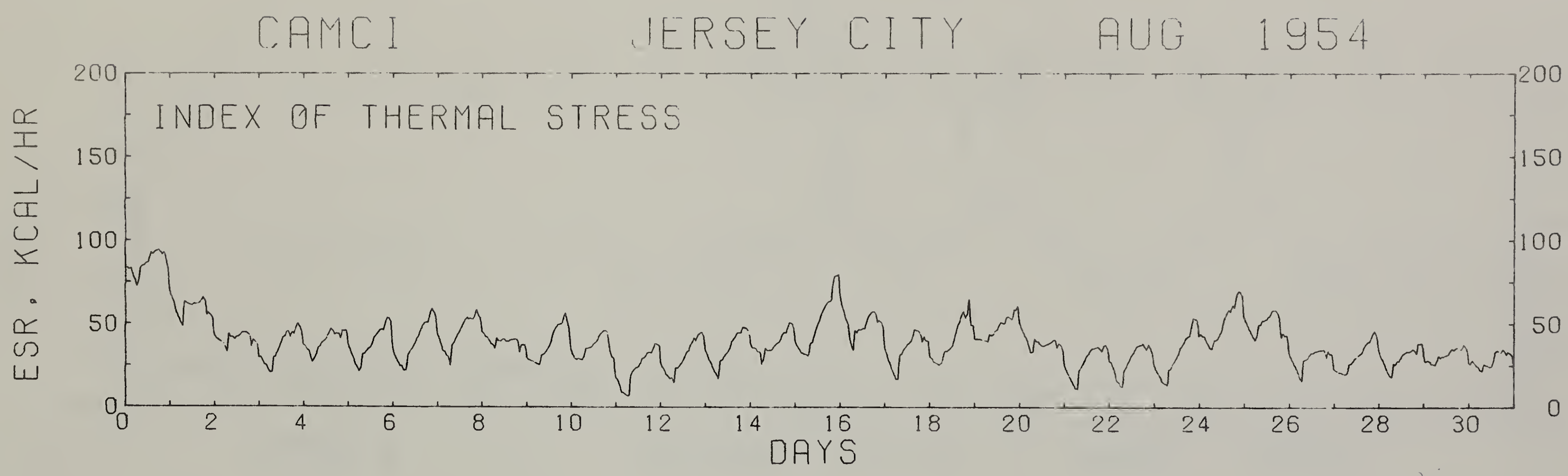
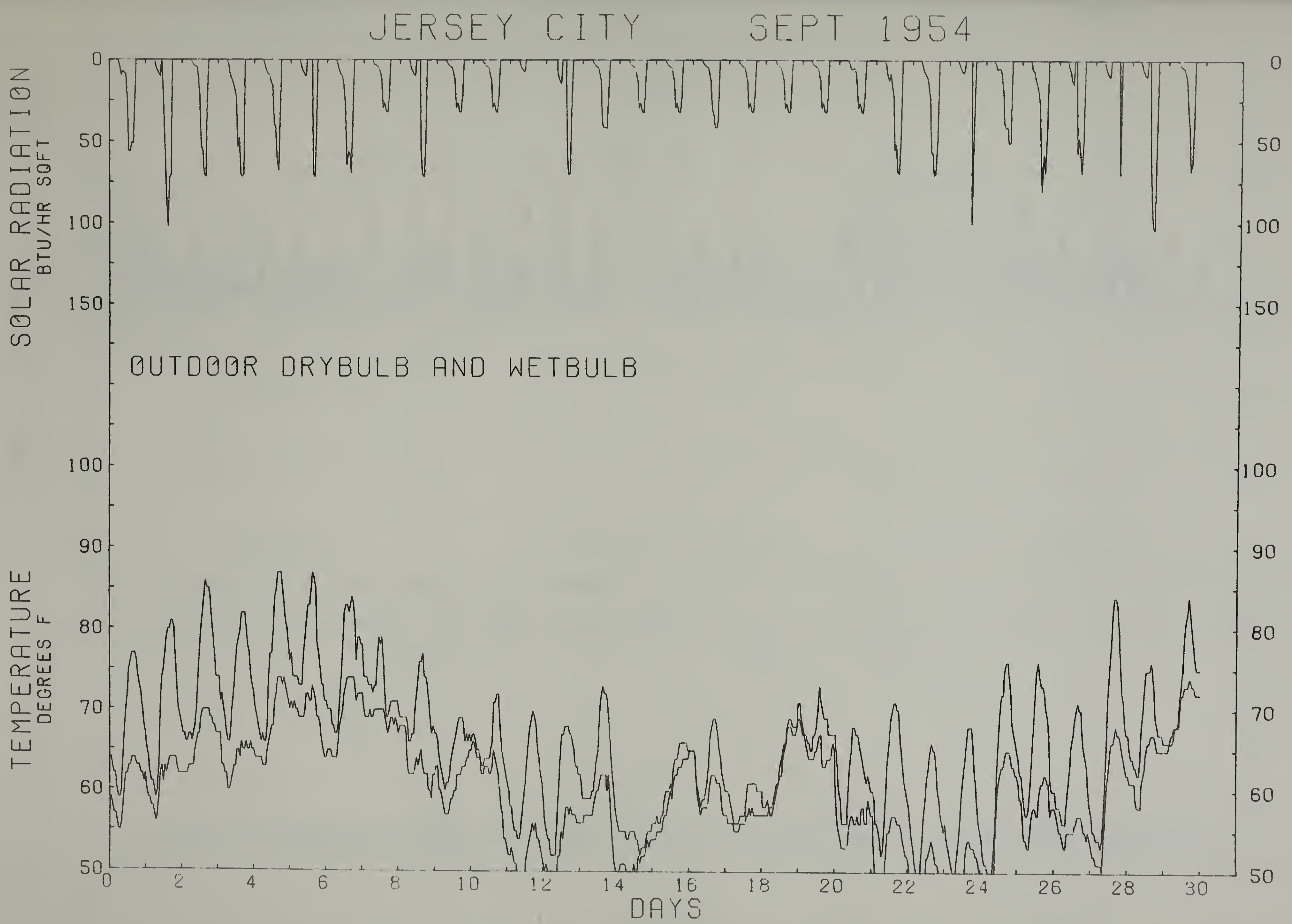

CAMCI
JERSEY CITY
AUG 1954
INDEX ØF THERMAL STRESS
ESR, KCAL/HR
DAYS
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
0
50
100
150
200

Figure 58



JERSEY CITY SEPT 1954
SOLAR RADIATION
BTU/HR SQFT
0
50
100
150
OUTDOOR DRYBULB AND WETBULB
TEMPERATURE
DEGREES F
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 59

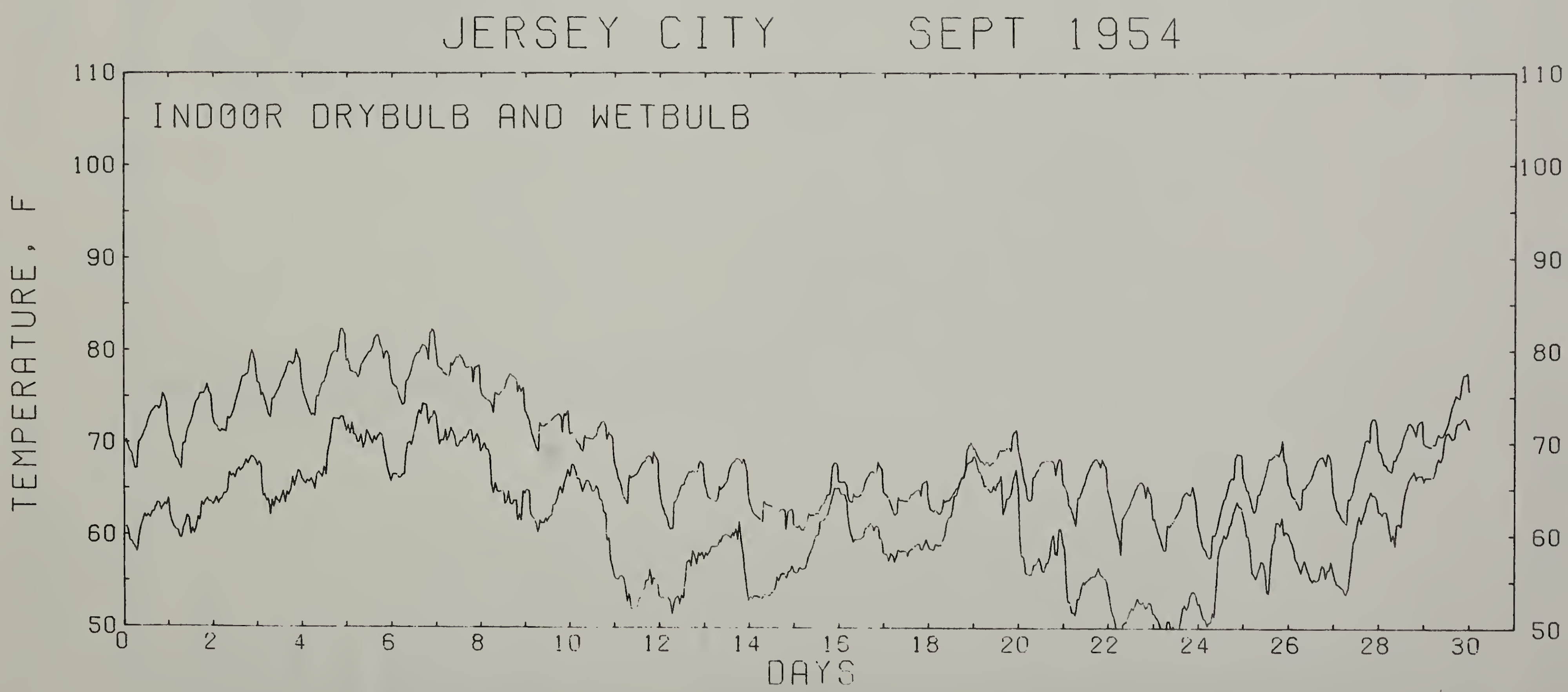
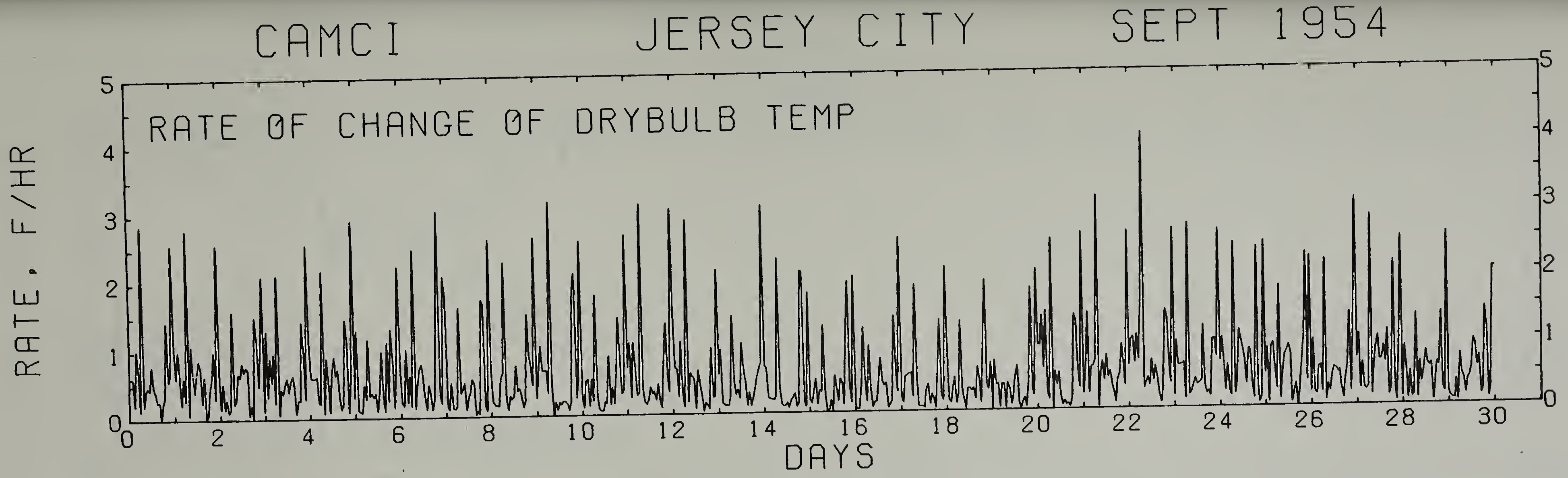

JERSEY CITY SEPT 1954
INDOOR DRYBULB AND WETBULB
TEMPERATURE, F
110
100
90
80
70
60
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 60


CAMCI
JERSEY CITY
SEPT 1954
RATE OF CHANGE OF DRYBULB TEMP
RATE, F/HR
DAYS
0
1
2
3
4
5
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

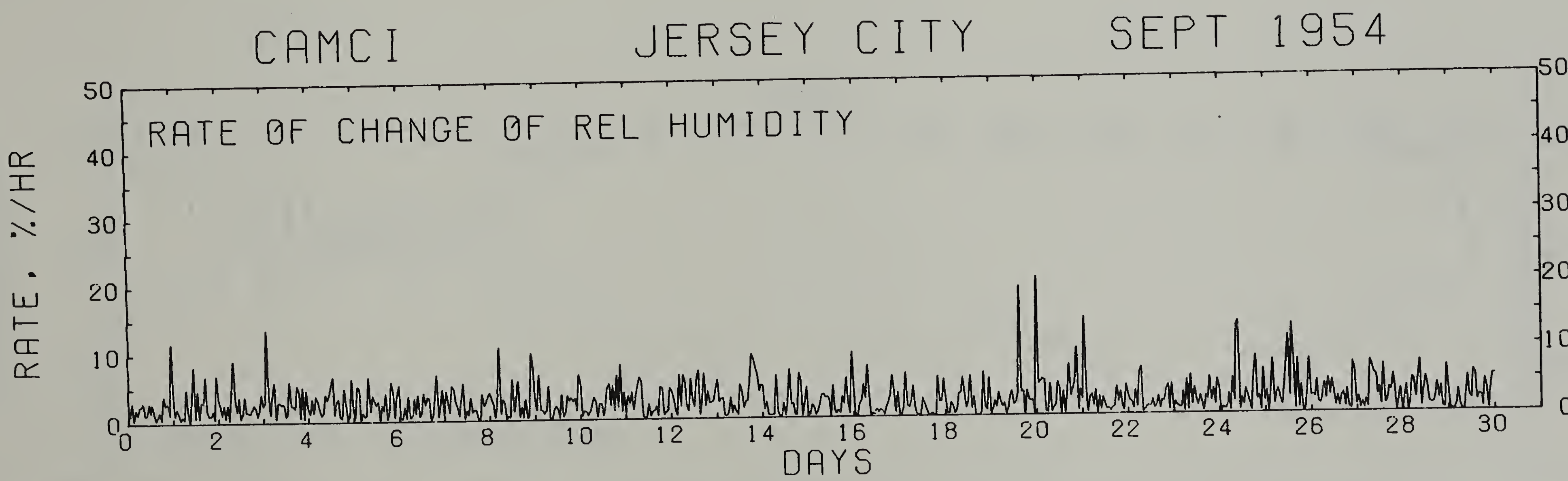
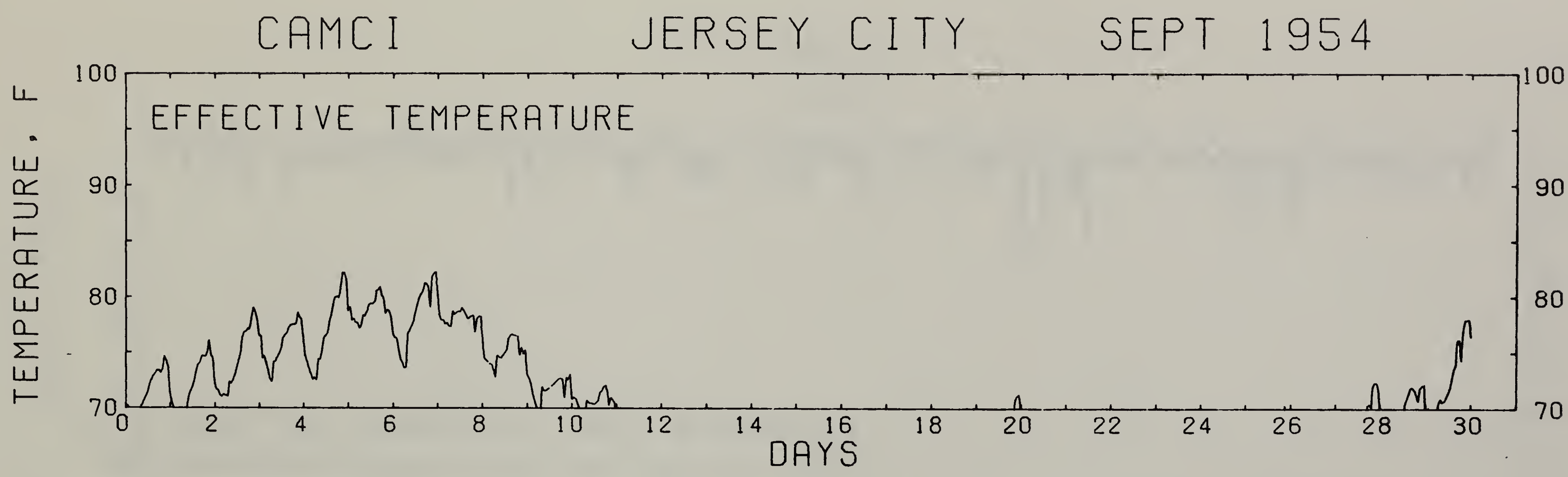

CAMCI
JERSEY CITY
SEPT 1954
RATE OF CHANGE OF REL HUMIDITY
RATE, %/HR
DAYS
0
10
20
30
40
50
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Figure 61


CAMCI
JERSEY CITY
SEPT 1954
EFFECTIVE TEMPERATURE
TEMPERATURE, F
DAYS

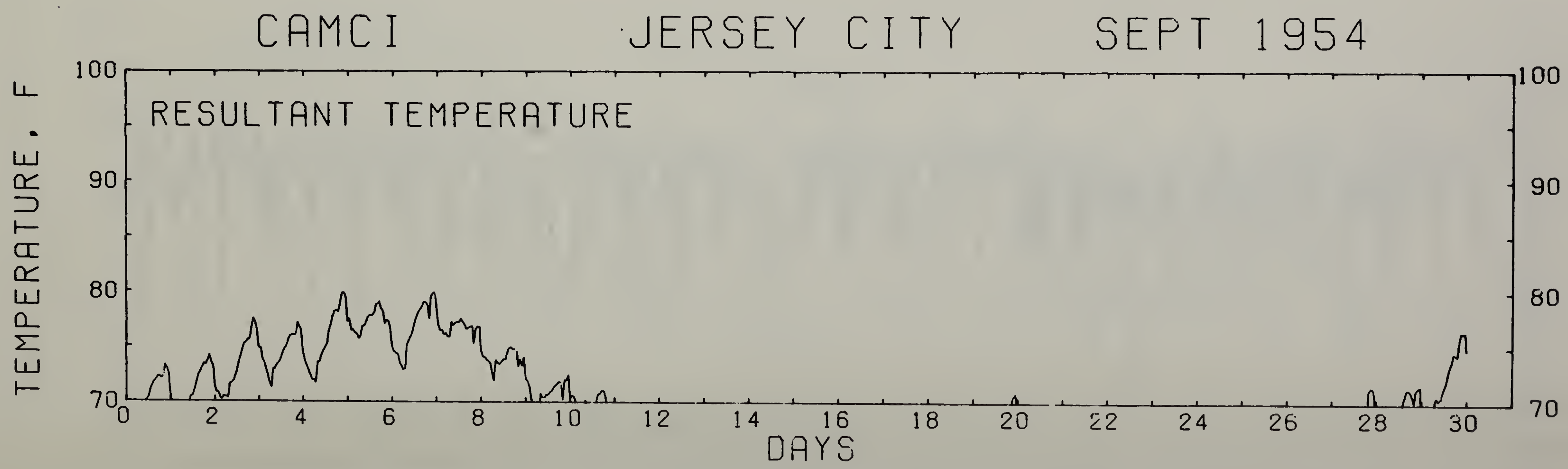
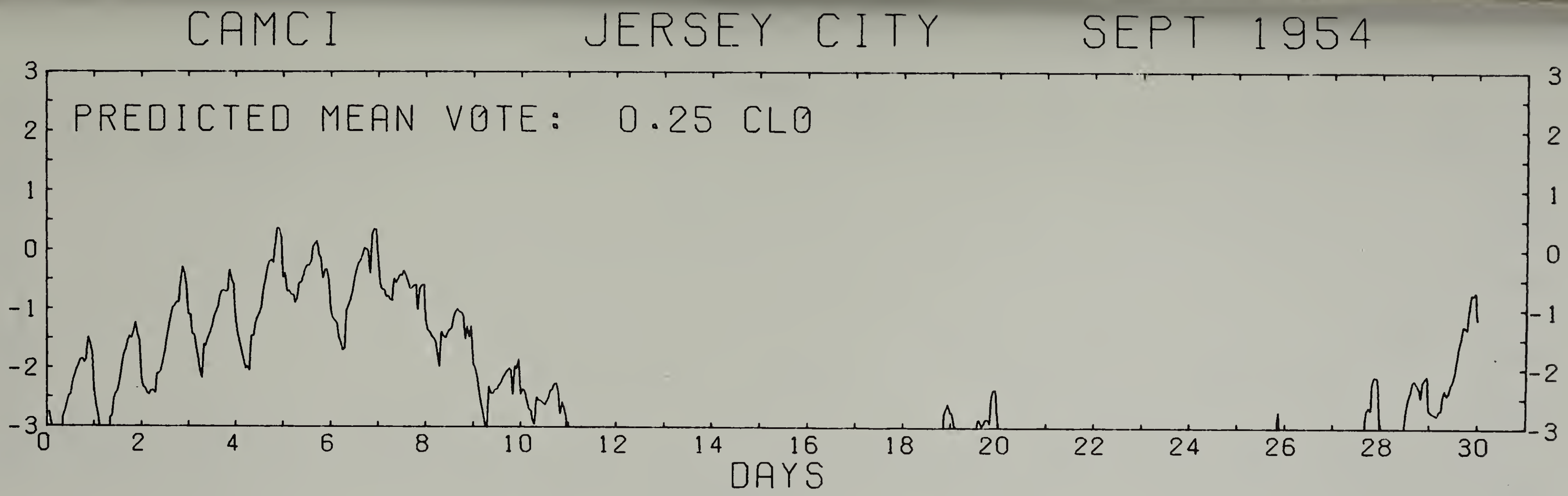

CAMCI
JERSEY CITY
SEPT 1954
RESULTANT TEMPERATURE
TEMPERATURE, F
DAYS

Figure 62


CAMCI
JERSEY CITY
SEPT 1954
PREDICTED MEAN VOTE: 0.25 CLO
DAYS

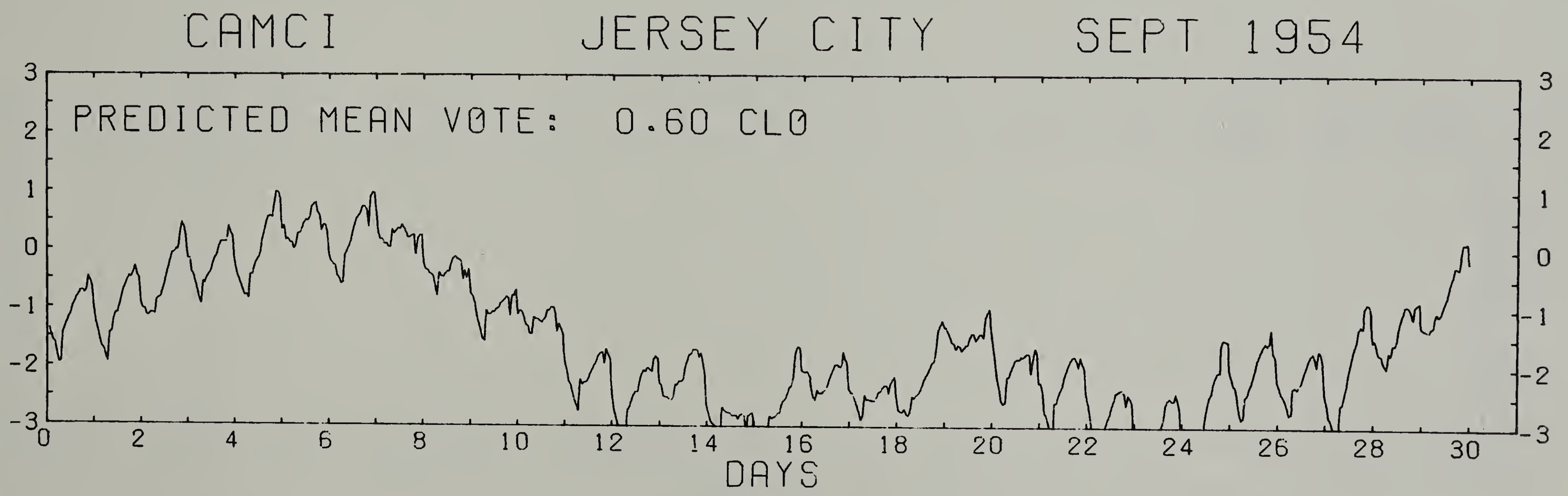

CAMCI
JERSEY CITY
SEPT 1954
PREDICTED MEAN VOTE: 0.60 CLO
DAYS

Figure 63

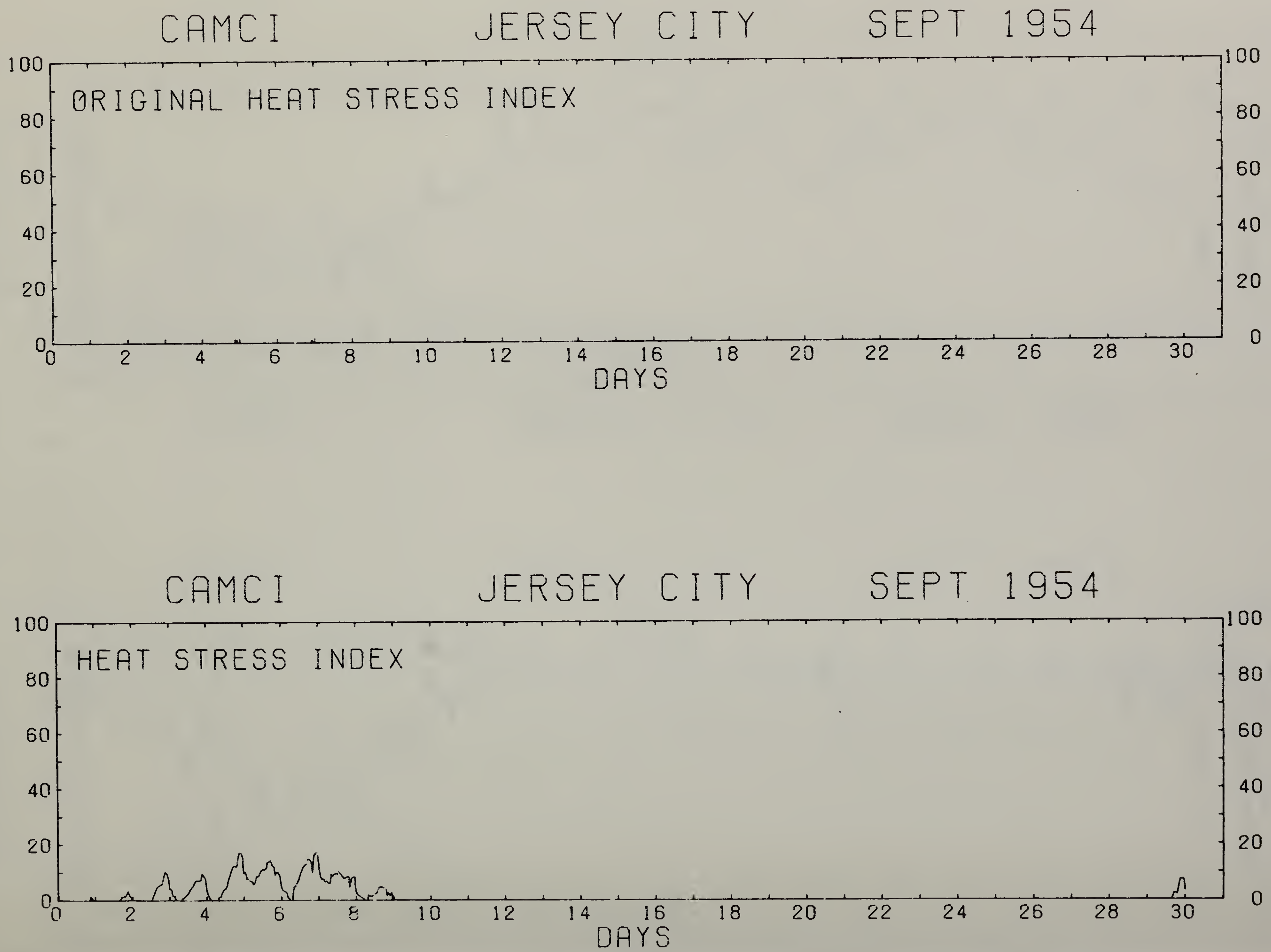
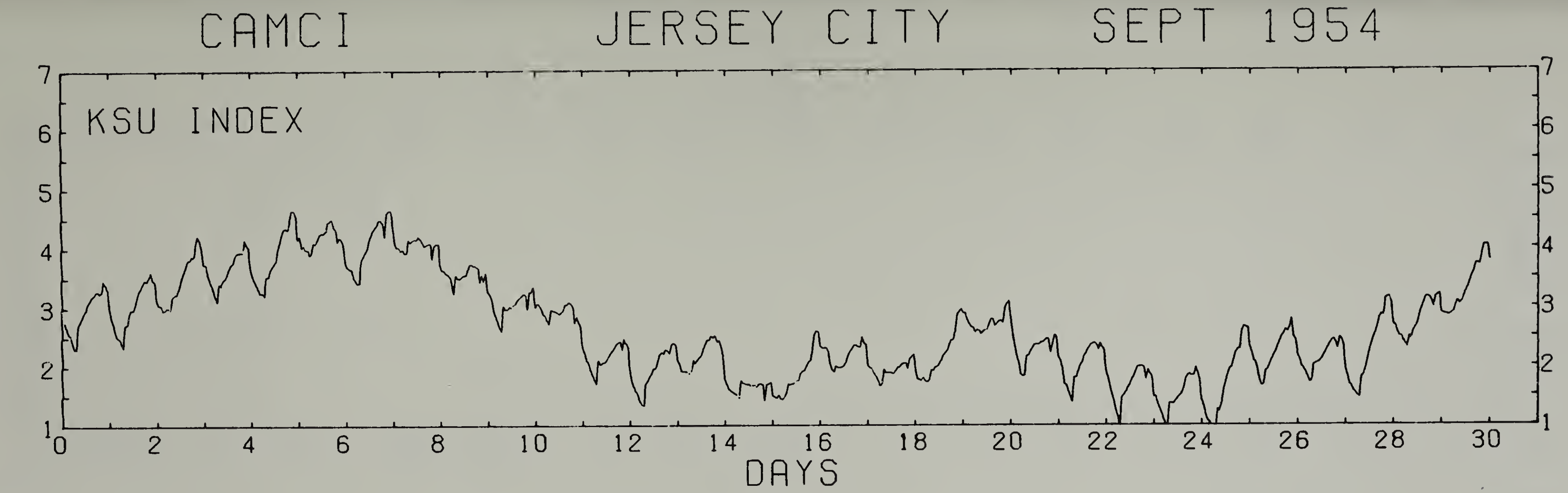

CAMCI
JERSEY CITY
SEPT 1954
ØRIGINAL HEAT STRESS INDEX
DAYS
CAMCI
JERSEY CITY
SEPT 1954
HEAT STRESS INDEX
DAYS

Figure 64


CAMCI JERSEY CITY SEPT 1954
KSU INDEX
DAYS

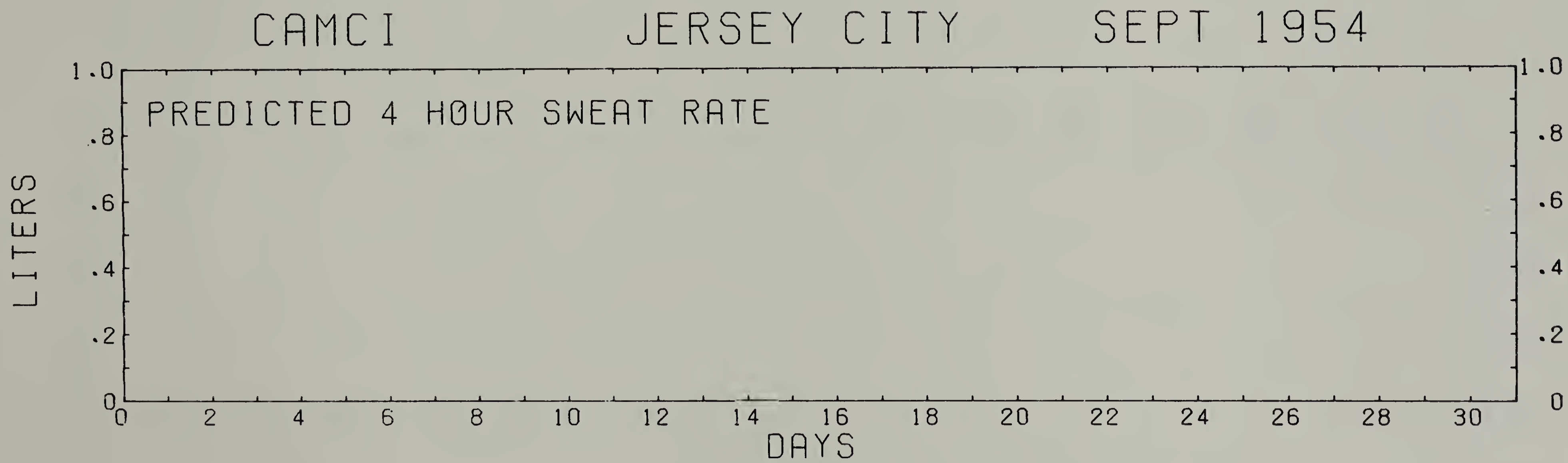

CAMCI JERSEY CITY SEPT 1954
PREDICTED 4 HOUR SWEAT RATE
LITERS
DAYS

Figure 65

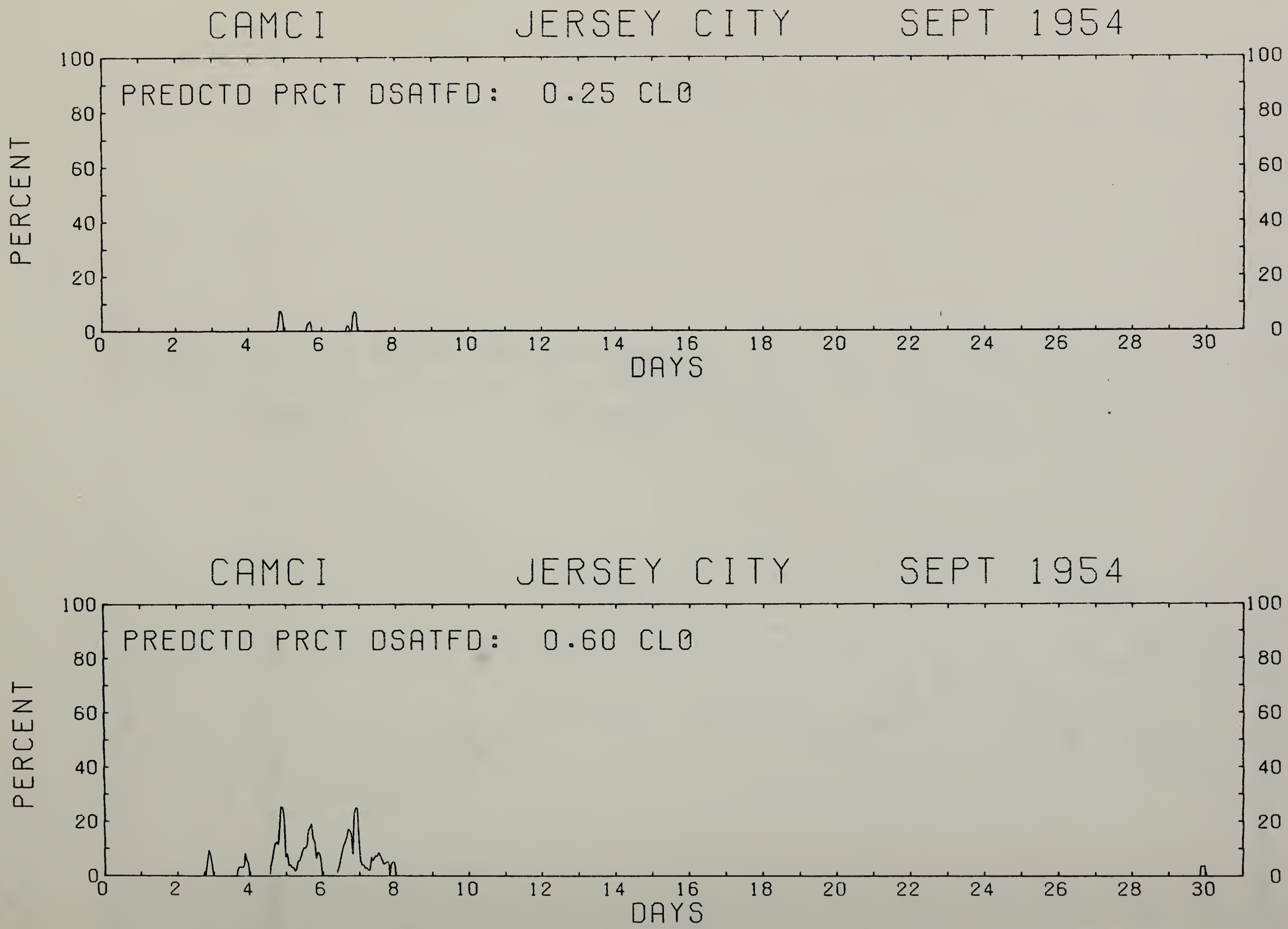

CAMCI
JERSEY CITY
SEPT 1954
PREDCTD PRCT DSATFD: 0.25 CLO
PERCENT
DAYS
CAMCI
JERSEY CITY
SEPT 1954
PREDCTD PRCT DSATFD: 0.60 CLO
PERCENT
DAYS

Figure 66

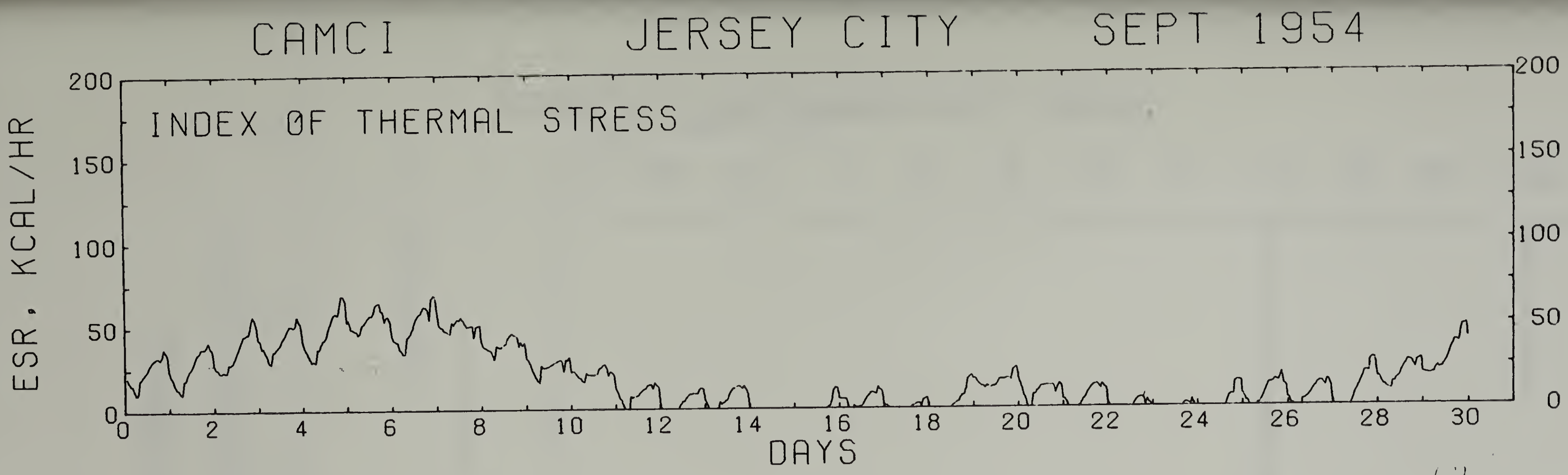

CAMCI
JERSEY CITY
SEPT 1954
INDEX ØF THERMAL STRESS
ESR, KCAL/HR
200
150
100
50
0
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
DAYS

Figure 67

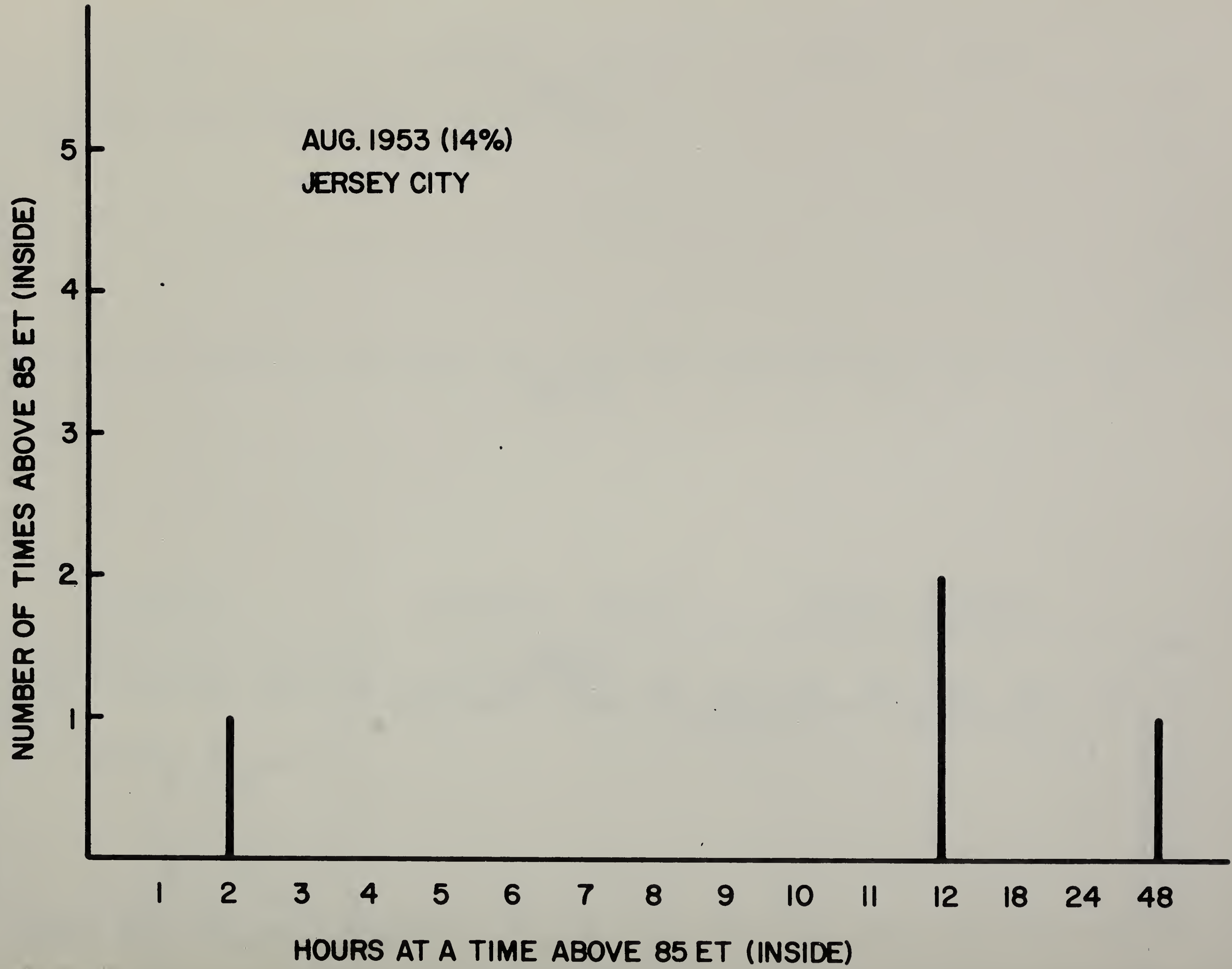

AUG. 1953 (14%)
JERSEY CITY
NUMBER OF TIMES ABOVE 85 ET (INSIDE)
5
4
3
2
1
1 2 3 4 5 6 7 8 9 10 11 12 18 24 48
HOURS AT A TIME ABOVE 85 ET (INSIDE)

Figure 68

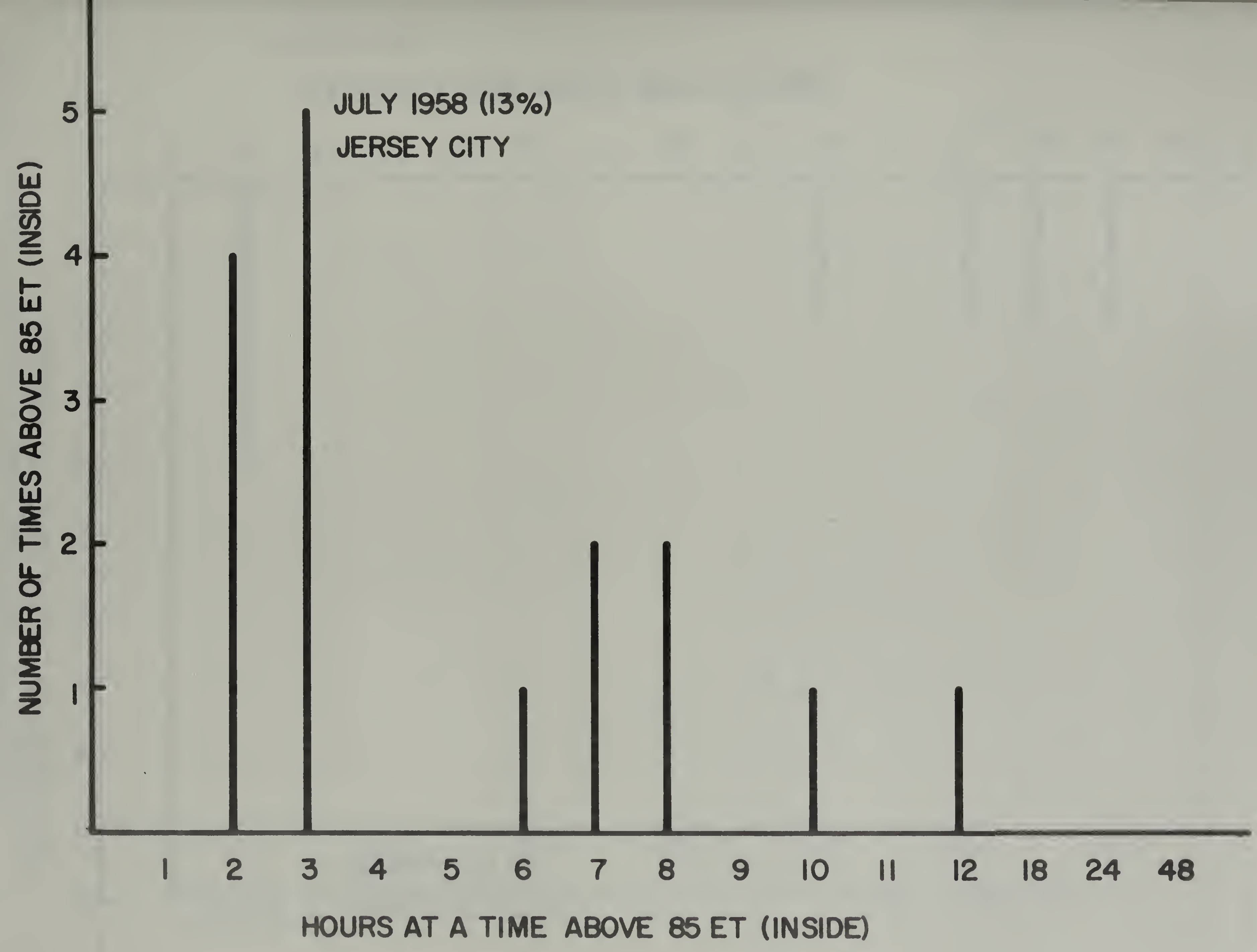

JULY 1958 (13%)
JERSEY CITY
NUMBER OF TIMES ABOVE 85 ET (INSIDE)
1
2
3
4
5
1 2 3 4 5 6 7 8 9 10 11 12 18 24 48
HOURS AT A TIME ABOVE 85 ET (INSIDE)

Figure 69

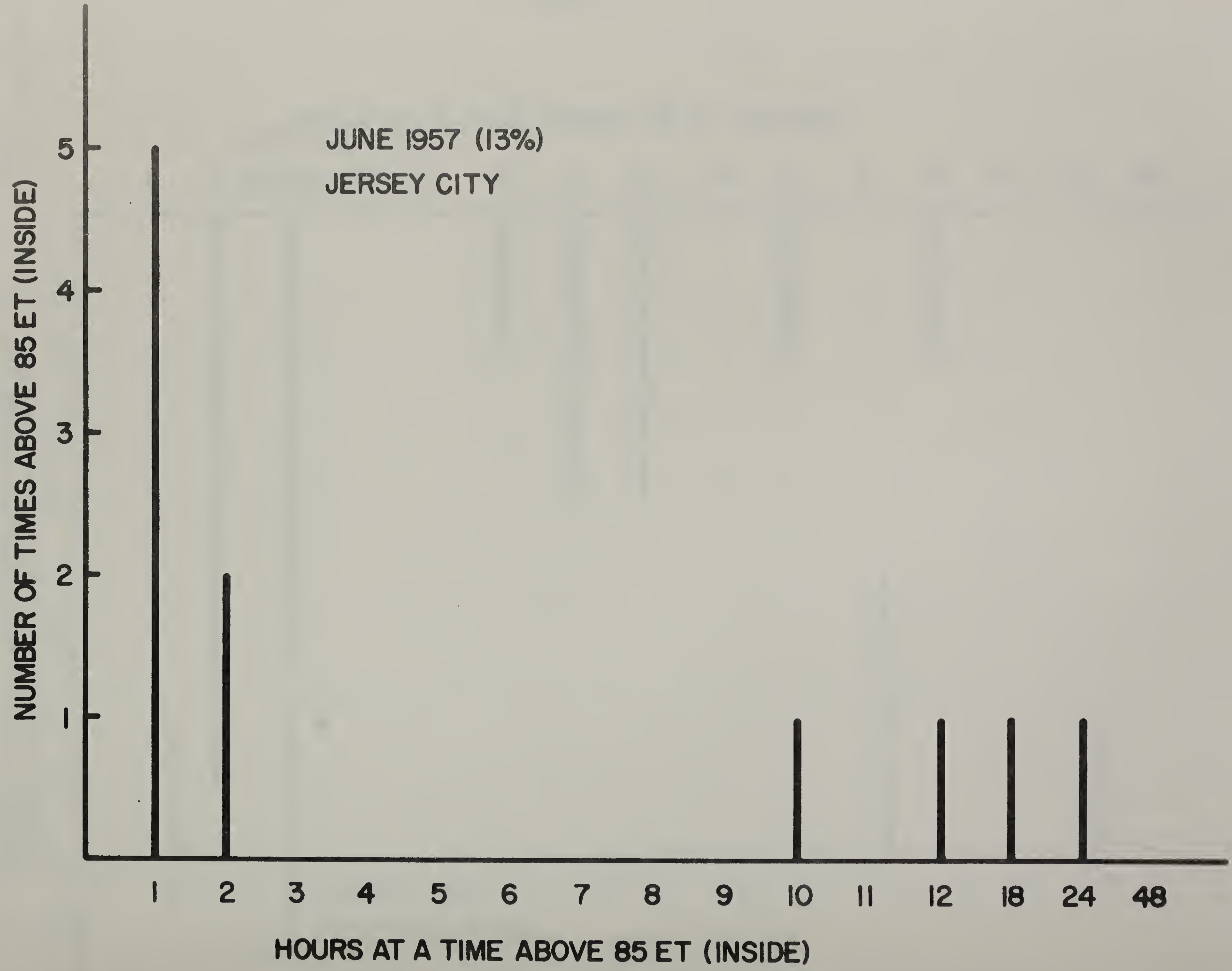

JUNE 1957 (13%)
JERSEY CITY
NUMBER OF TIMES ABOVE 85 ET (INSIDE)
5
4
3
2
1
1 2 3 4 5 6 7 8 9 10 11 12 18 24 48
HOURS AT A TIME ABOVE 85 ET (INSIDE)

Figure 70

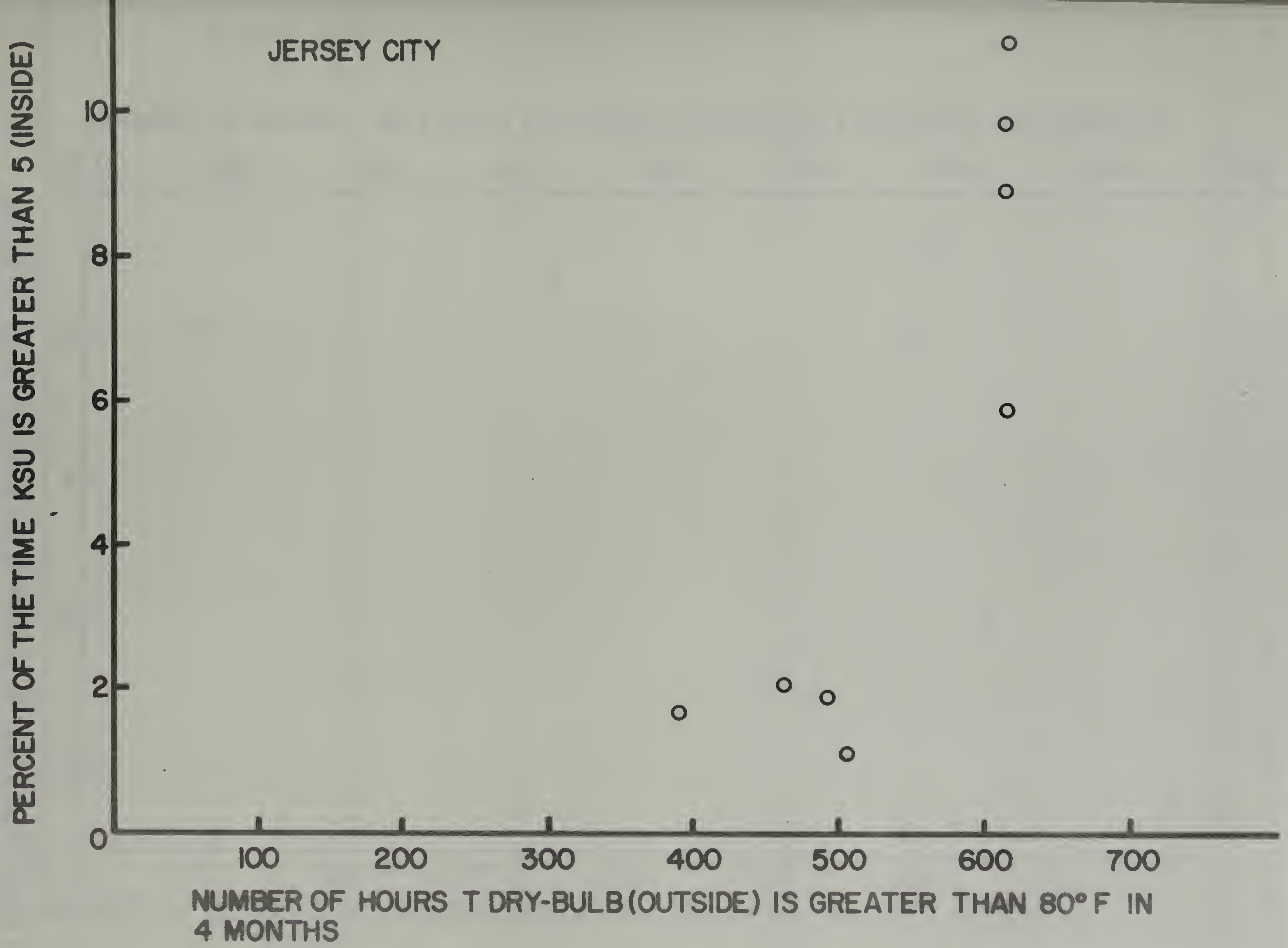

JERSEY CITY
PERCENT OF THE TIME KSU IS GREATER THAN 5 (INSIDE)
10
8
6
4
2
0
100
200
300
400
500
600
700
NUMBER OF HOURS T DRY-BULB (OUTSIDE) IS GREATER THAN 80° F IN 4 MONTHS

Figure 71

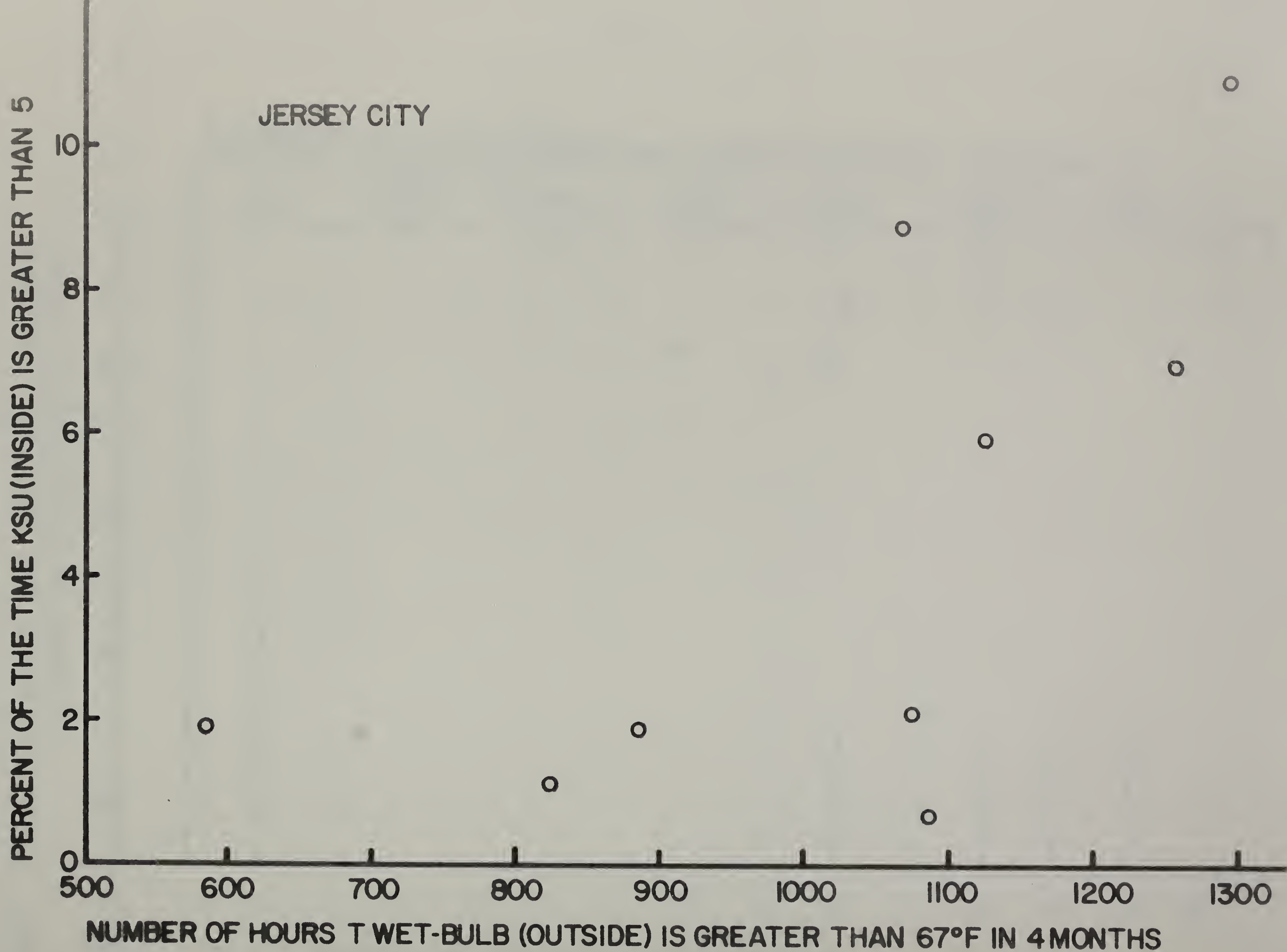

JERSEY CITY
PERCENT OF THE TIME KSU (INSIDE) IS GREATER THAN 5
0
2
4
6
8
10
500
600
700
800
900
1000
1100
1200
1300
NUMBER OF HOURS T WET-BULB (OUTSIDE) IS GREATER THAN 67°F IN 4 MONTHS

Figure 72

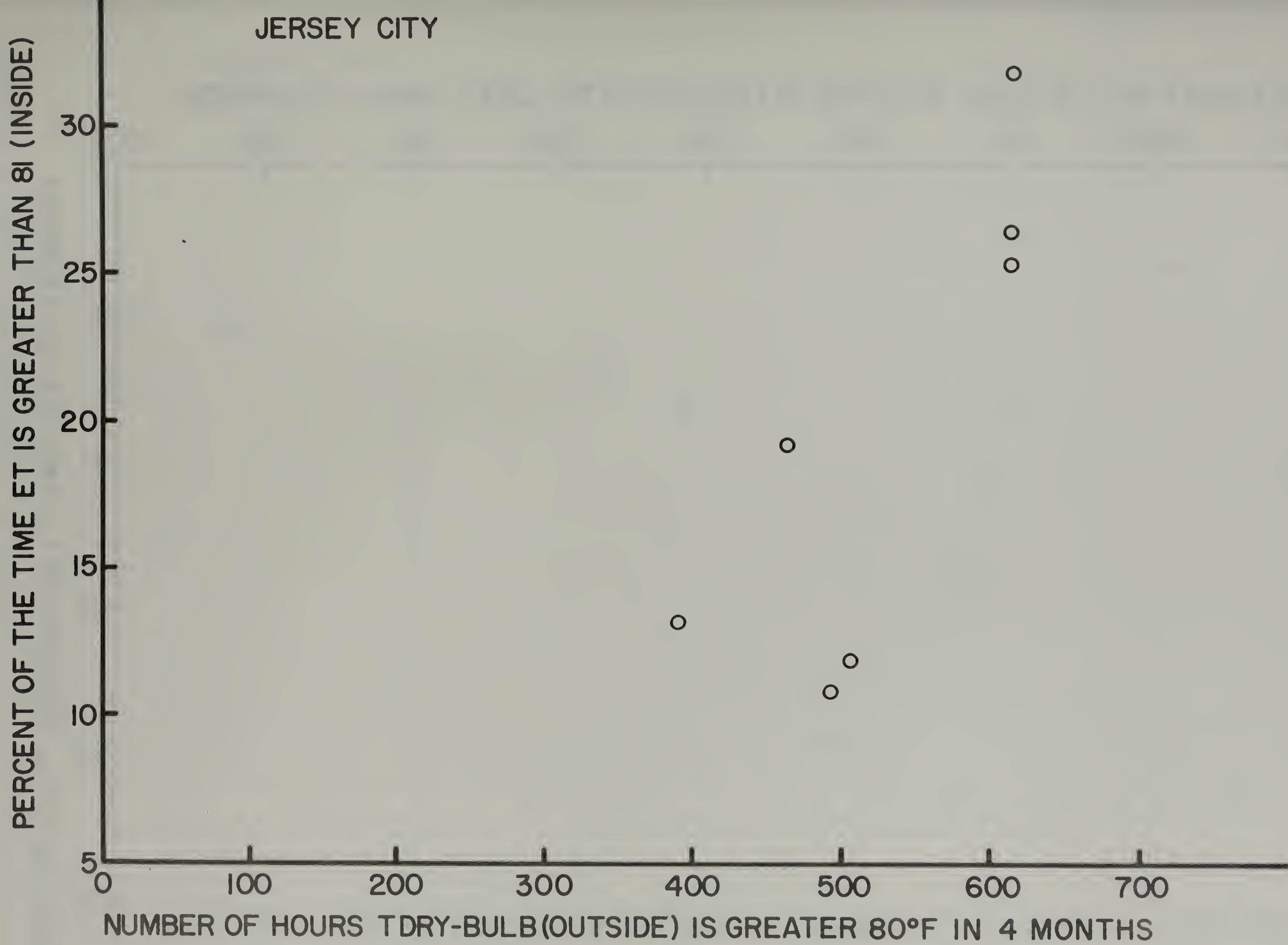

JERSEY CITY
PERCENT OF THE TIME ET IS GREATER THAN 81 (INSIDE)
30
25
20
15
10
5
0
100
200
300
400
500
600
700
NUMBER OF HOURS T DRY-BULB (OUTSIDE) IS GREATER 80°F IN 4 MONTHS

Figure 73

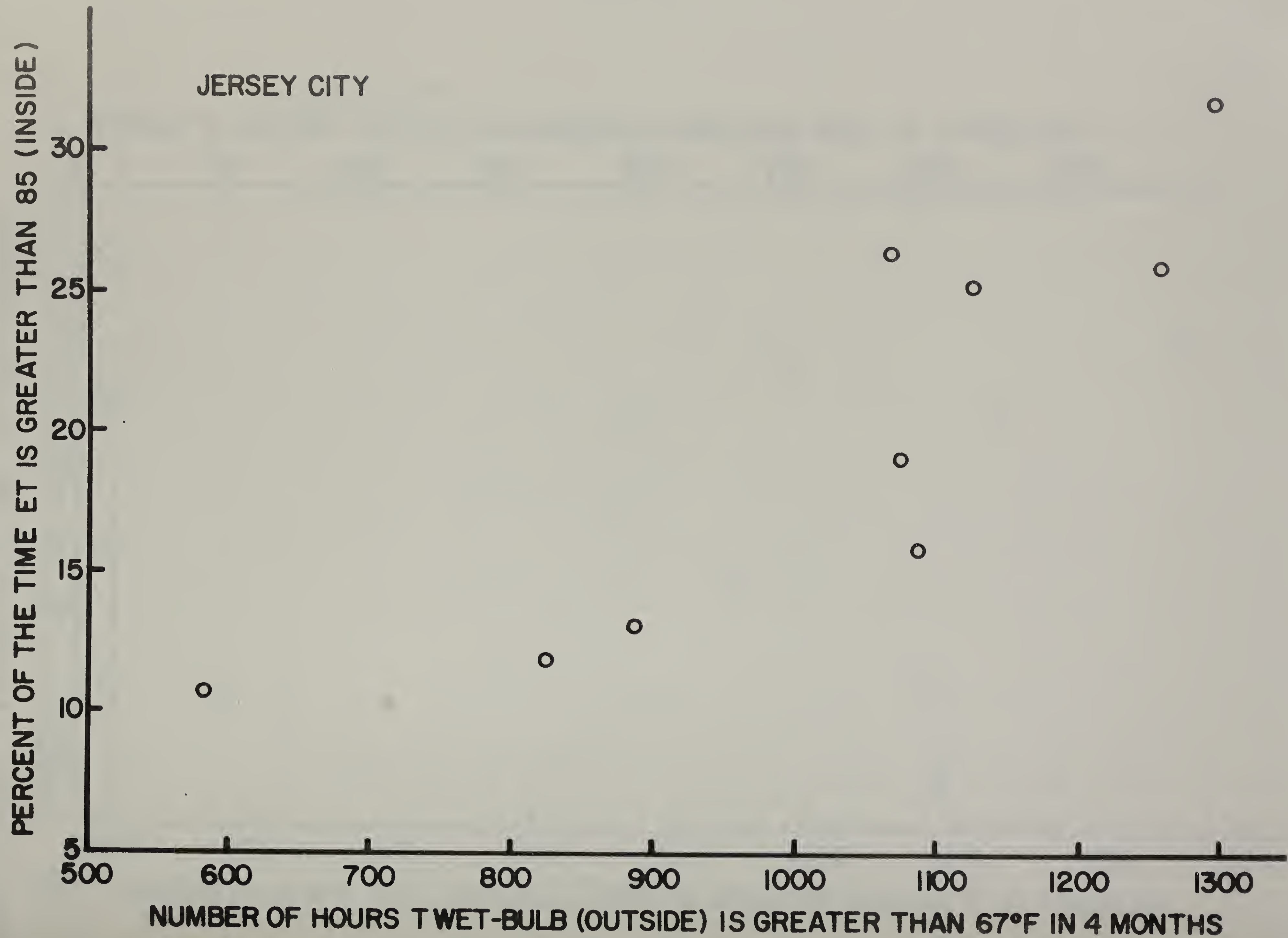

JERSEY CITY
PERCENT OF THE TIME ET IS GREATER THAN 85 (INSIDE)
30
25
20
15
10
5
500
600
700
800
900
1000
1100
1200
1300
NUMBER OF HOURS T WET-BULB (OUTSIDE) IS GREATER THAN 67°F IN 4 MONTHS

Figure 74

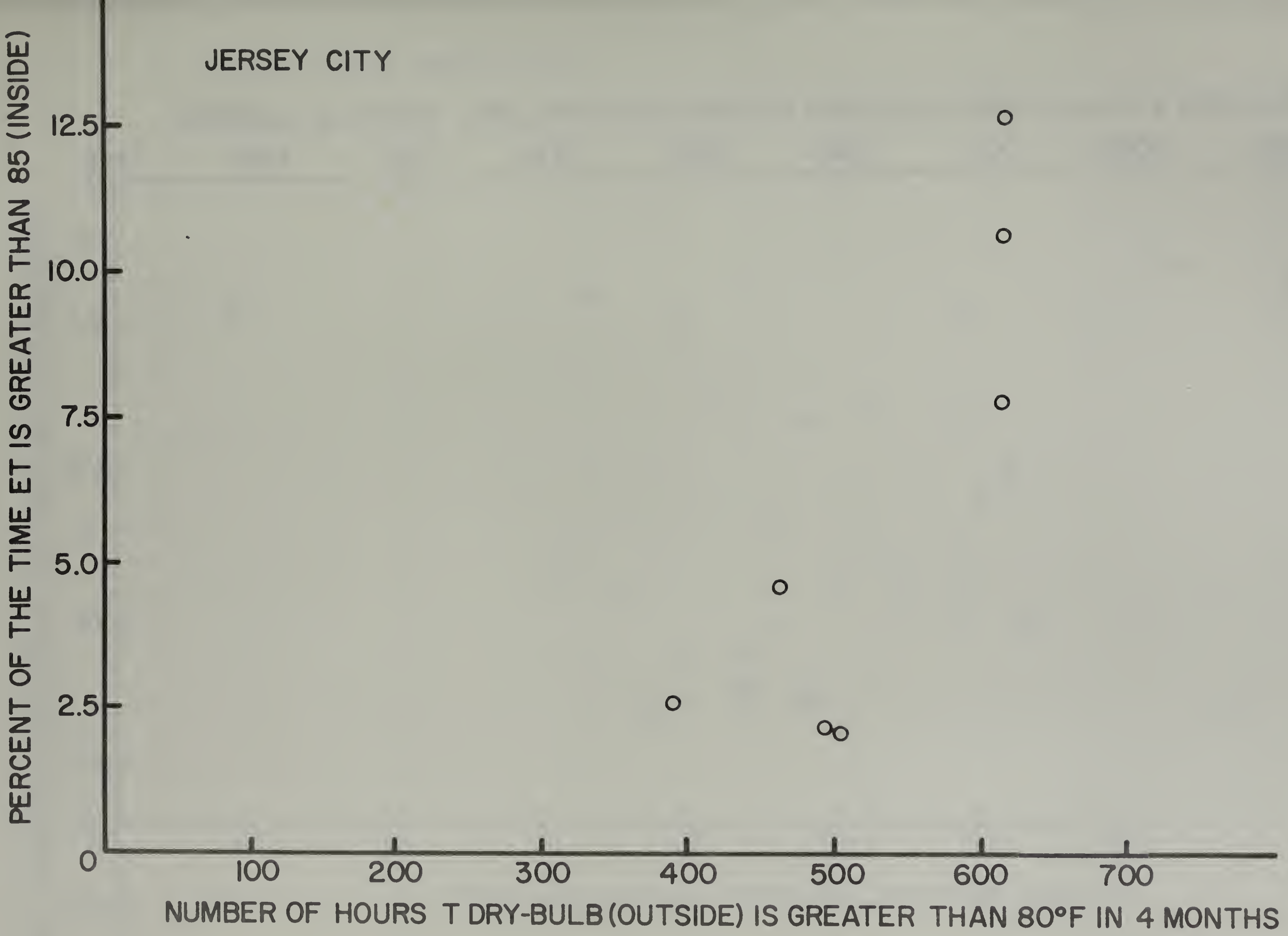

JERSEY CITY
PERCENT OF THE TIME ET IS GREATER THAN 85 (INSIDE)
12.5
10.0
7.5
5.0
2.5
0
100
200
300
400
500
600
700
NUMBER OF HOURS T DRY-BULB (OUTSIDE) IS GREATER THAN 80°F IN 4 MONTHS

Figure 75

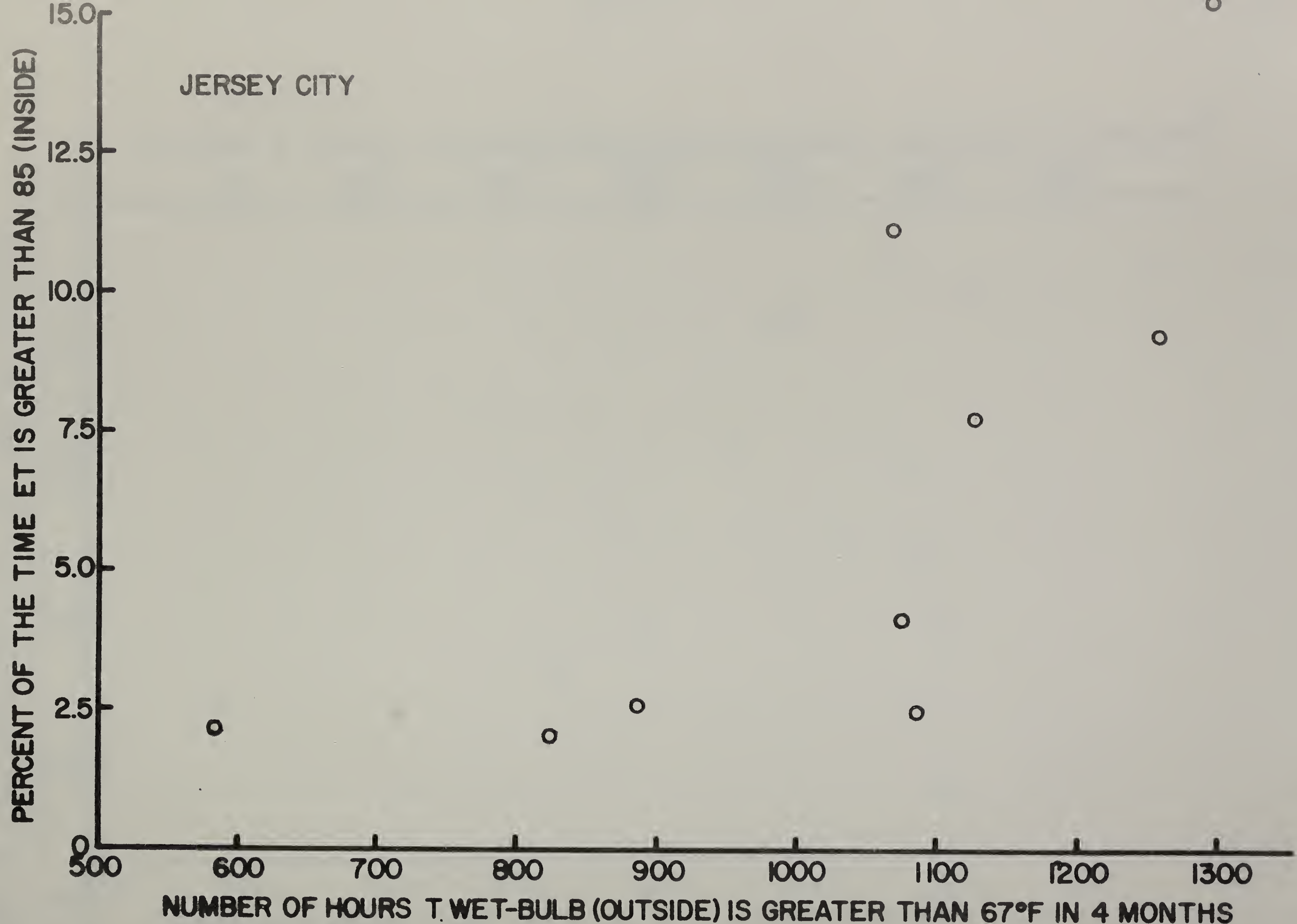

JERSEY CITY
PERCENT OF THE TIME ET IS GREATER THAN 85 (INSIDE)
15.0
12.5
10.0
7.5
5.0
2.5
0
500
600
700
800
900
1000
1100
1200
1300
NUMBER OF HOURS T. WET-BULB (OUTSIDE) IS GREATER THAN 67°F IN 4 MONTHS

Figure 76

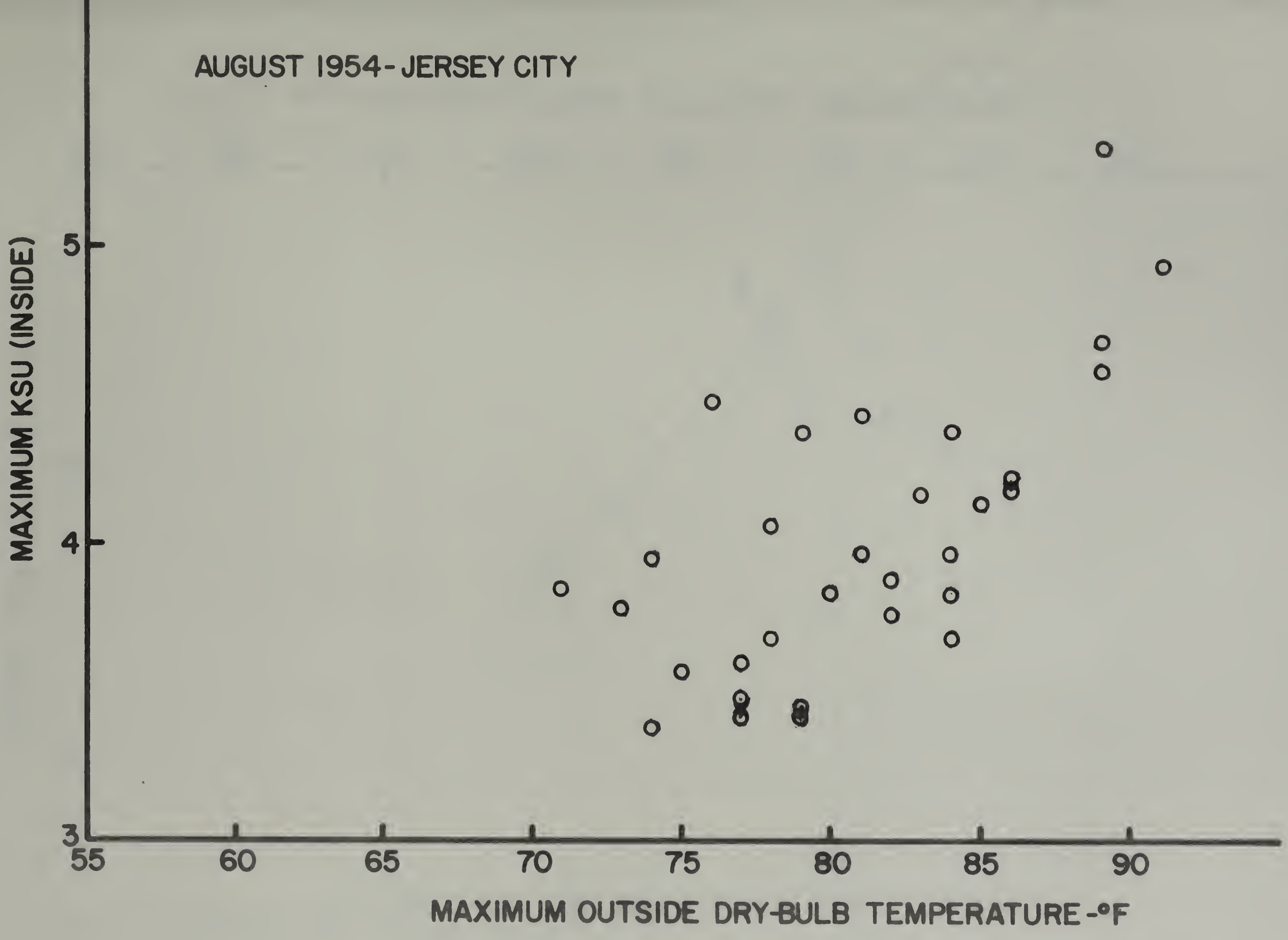

AUGUST 1954-JERSEY CITY
MAXIMUM KSU (INSIDE)
5
4
3
55
60
65
70
75
80
85
90
MAXIMUM OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 77

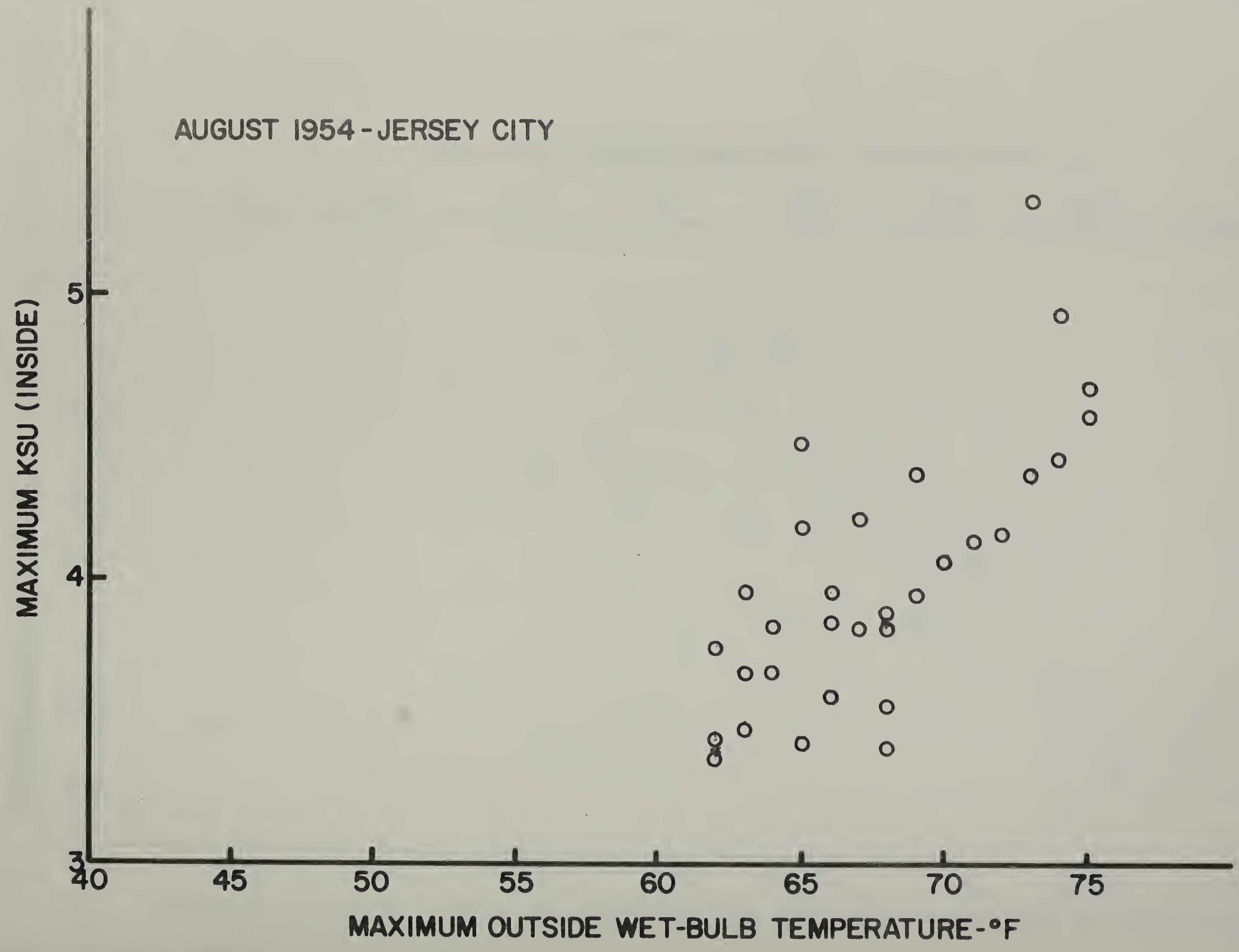

AUGUST 1954 - JERSEY CITY
MAXIMUM KSU (INSIDE)
5
4
3
40
45
50
55
60
65
70
75
MAXIMUM OUTSIDE WET-BULB TEMPERATURE - °F

Figure 78

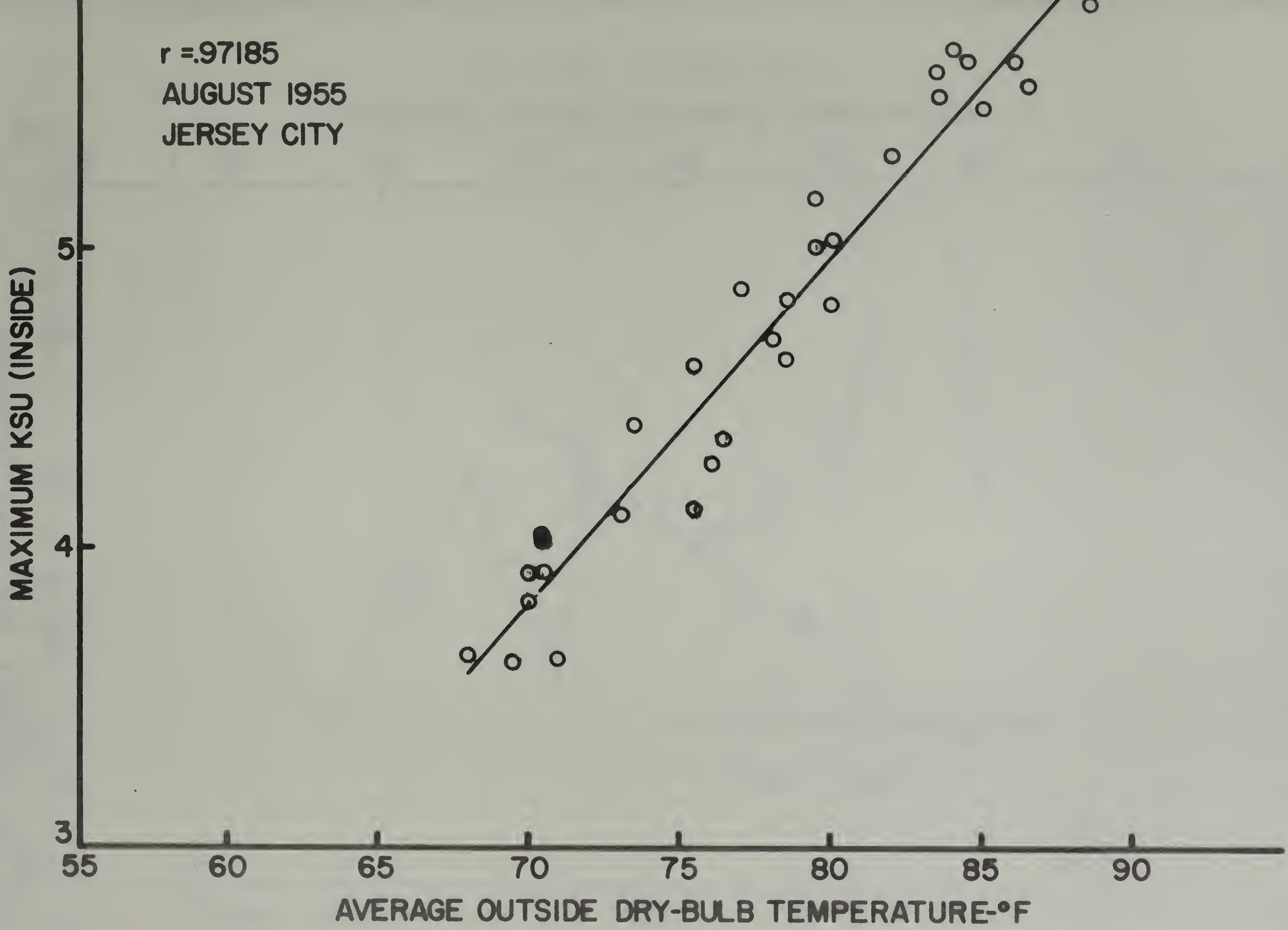

r =.97185
AUGUST 1955
JERSEY CITY
MAXIMUM KSU (INSIDE)
5
4
3
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 79

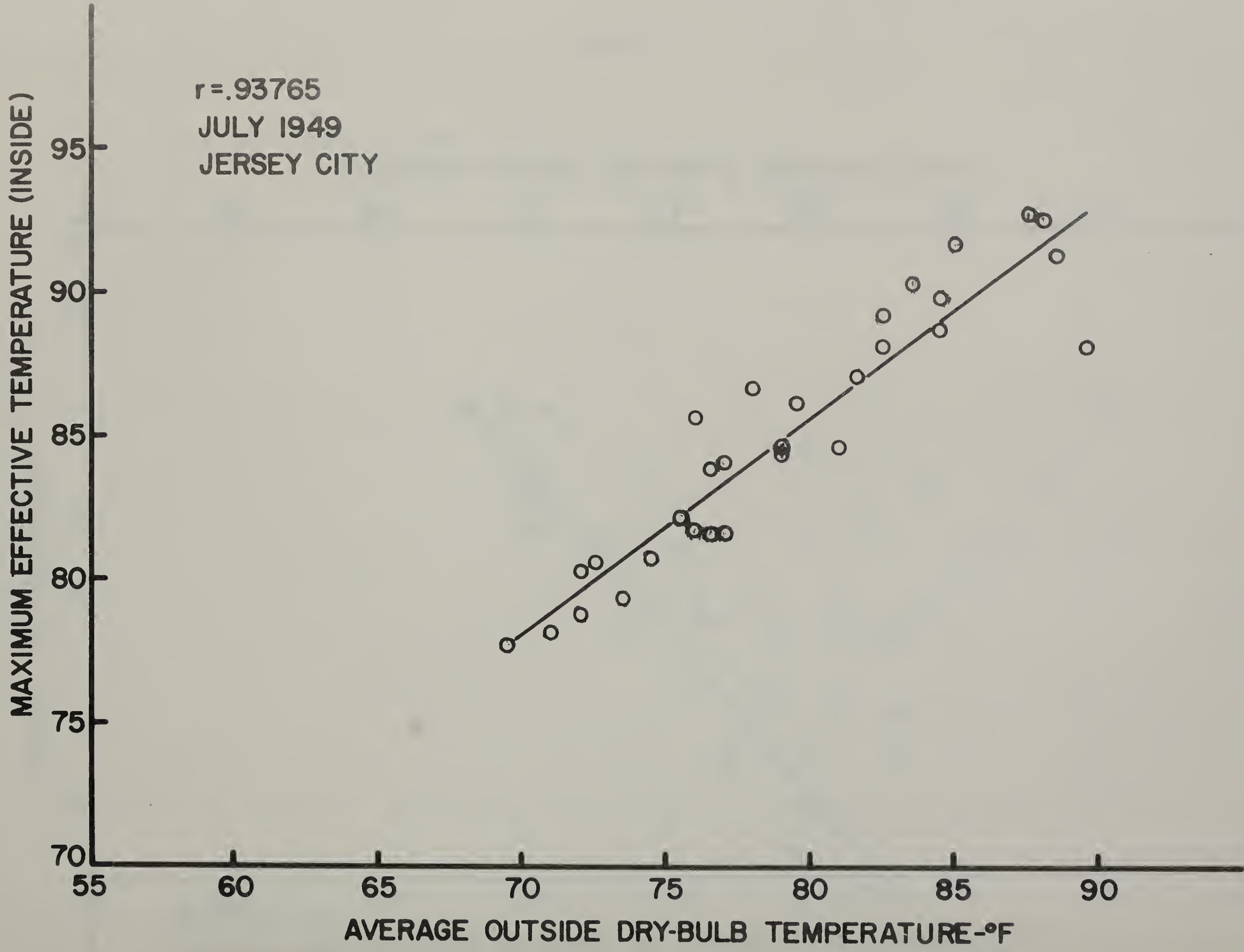

r=.93765
JULY 1949
JERSEY CITY
MAXIMUM EFFECTIVE TEMPERATURE (INSIDE)
95
90
85
80
75
70
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 80

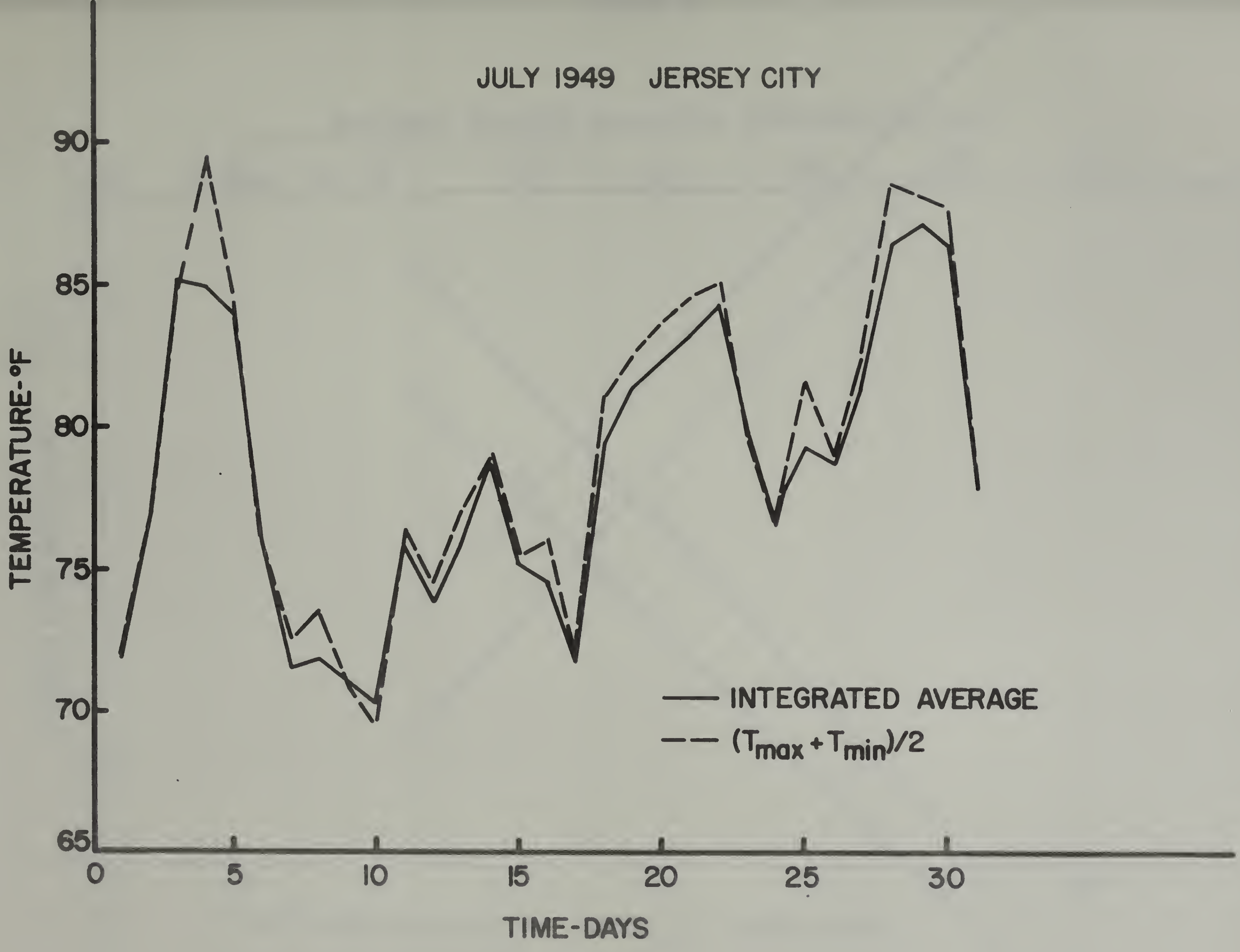

JULY 1949 JERSEY CITY
TEMPERATURE-°F
90
85
80
75
70
65
0
5
10
15
20
25
30
TIME-DAYS
INTEGRATED AVERAGE
$(T_{max} + T_{min})/2$

Figure 81

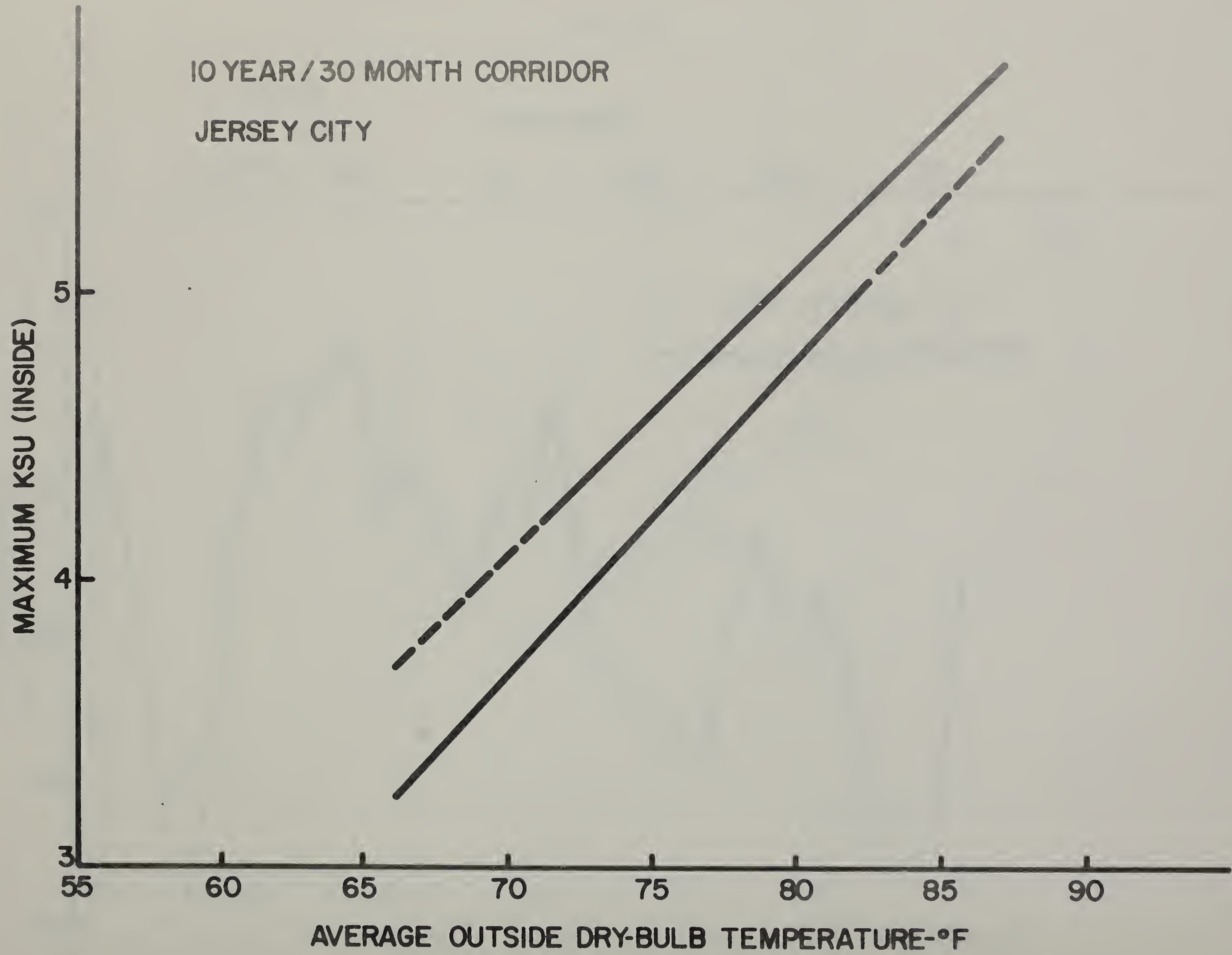

10 YEAR / 30 MONTH CORRIDOR
JERSEY CITY
MAXIMUM KSU (INSIDE)
3
4
5
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 82

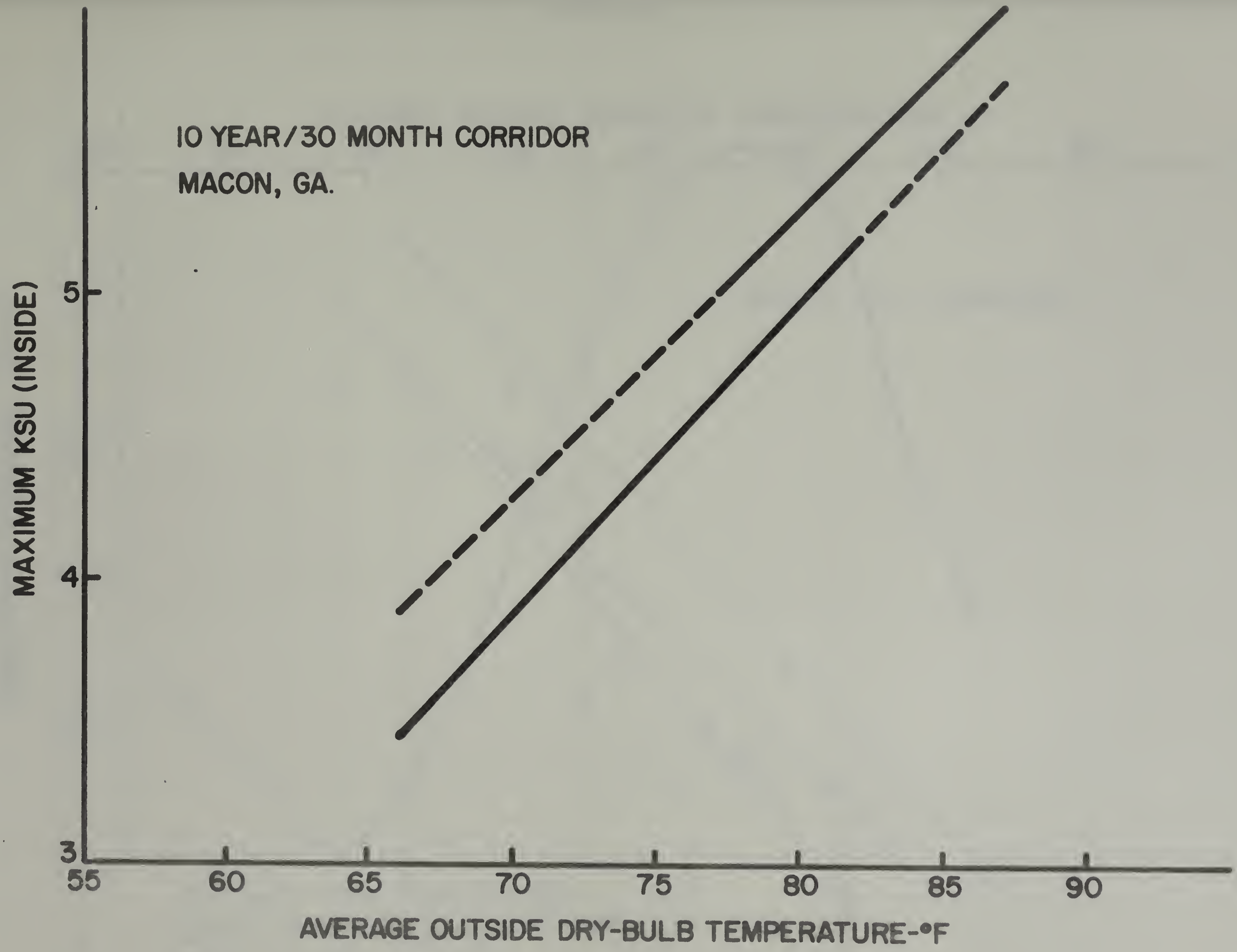

10 YEAR/30 MONTH CORRIDOR
MACON, GA.
MAXIMUM KSU (INSIDE)
3
4
5
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 83

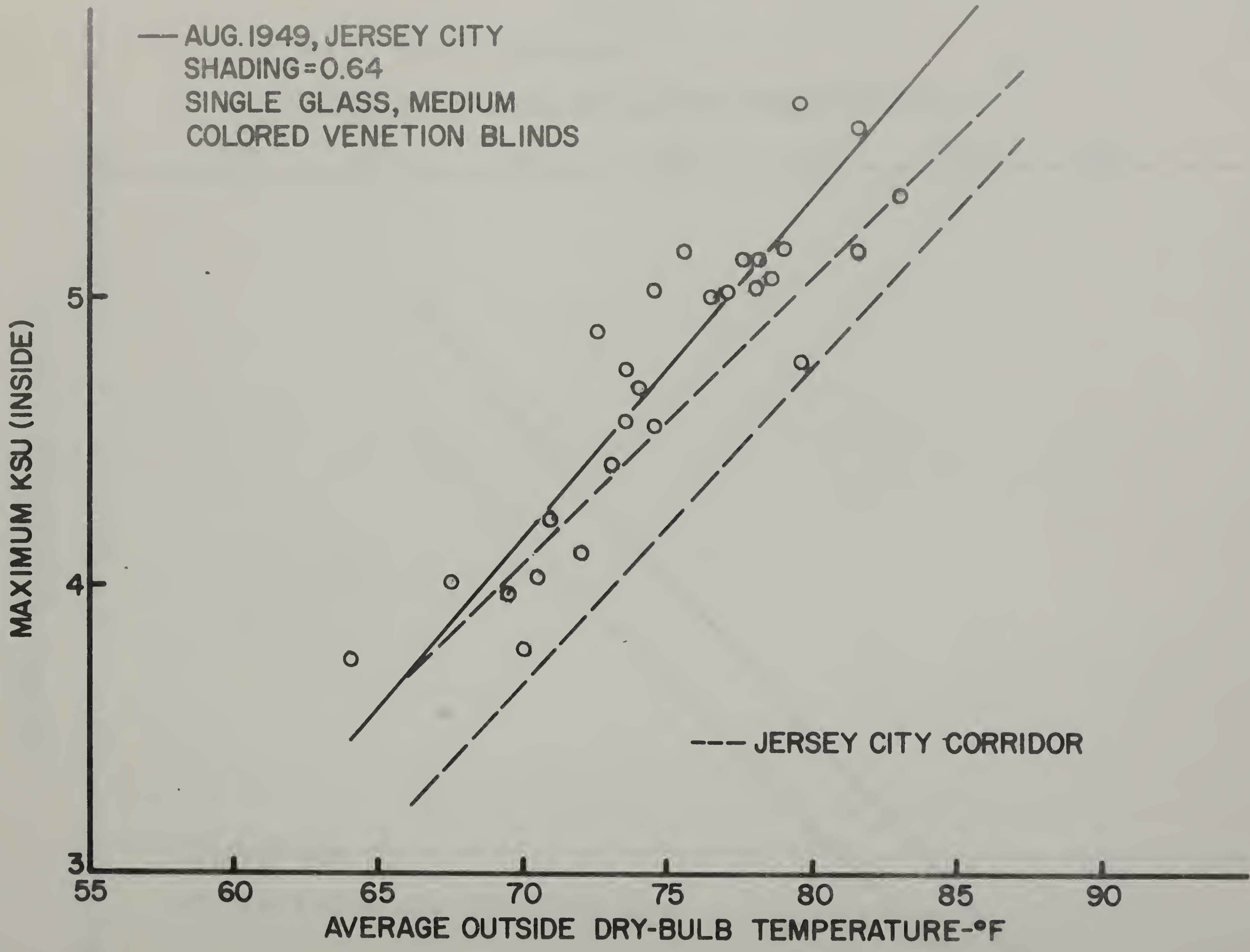
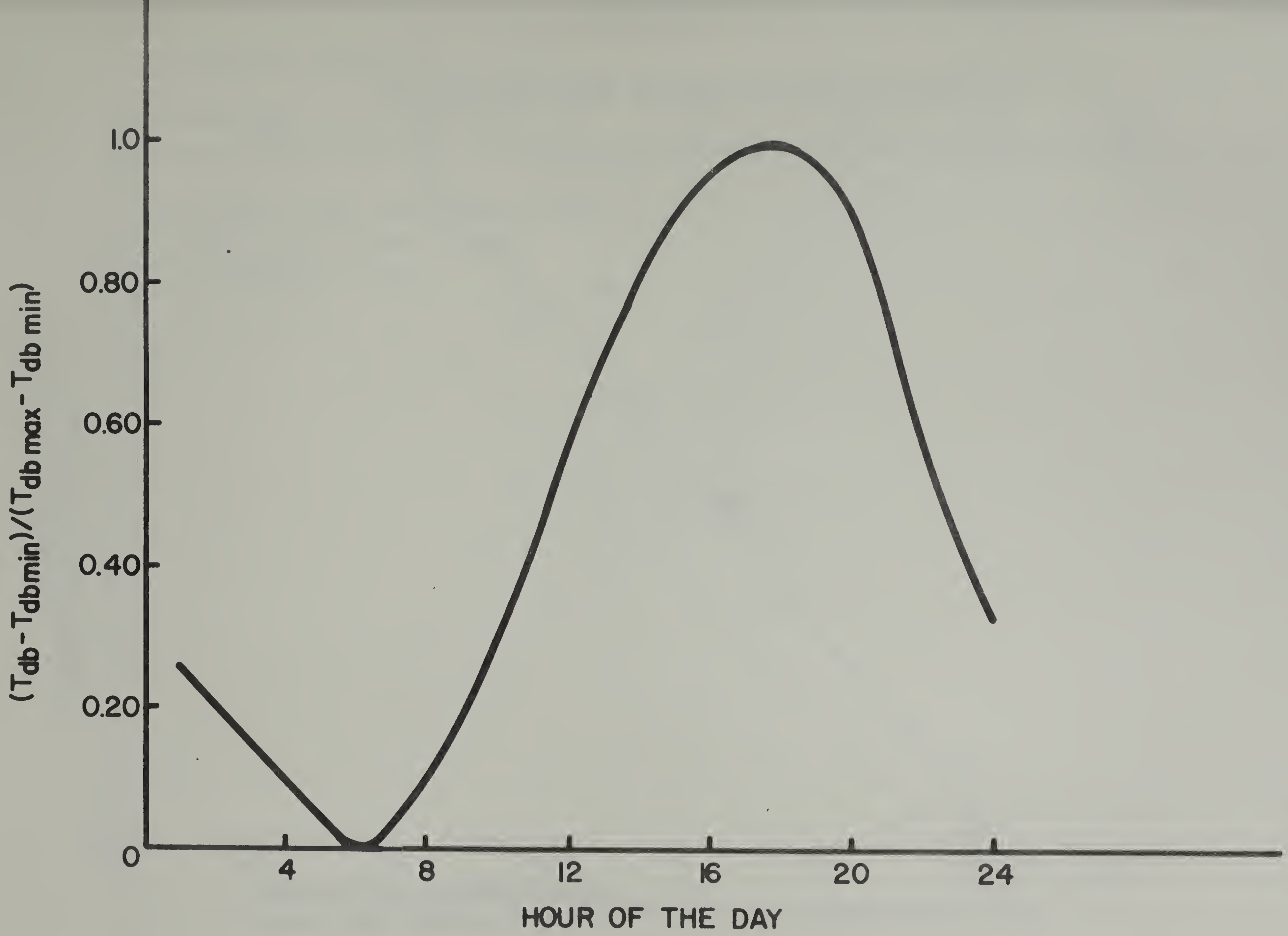

— AUG. 1949, JERSEY CITY
SHADING=0.64
SINGLE GLASS, MEDIUM
COLORED VENETION BLINDS
--- JERSEY CITY CORRIDOR
MAXIMUM KSU (INSIDE)
3
4
5
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 84



(Tdb - Tdbmin)/(Tdb max - Tdb min)
1.0
0.80
0.60
0.40
0.20
0
4
8
12
16
20
24
HOUR OF THE DAY

Figure 86

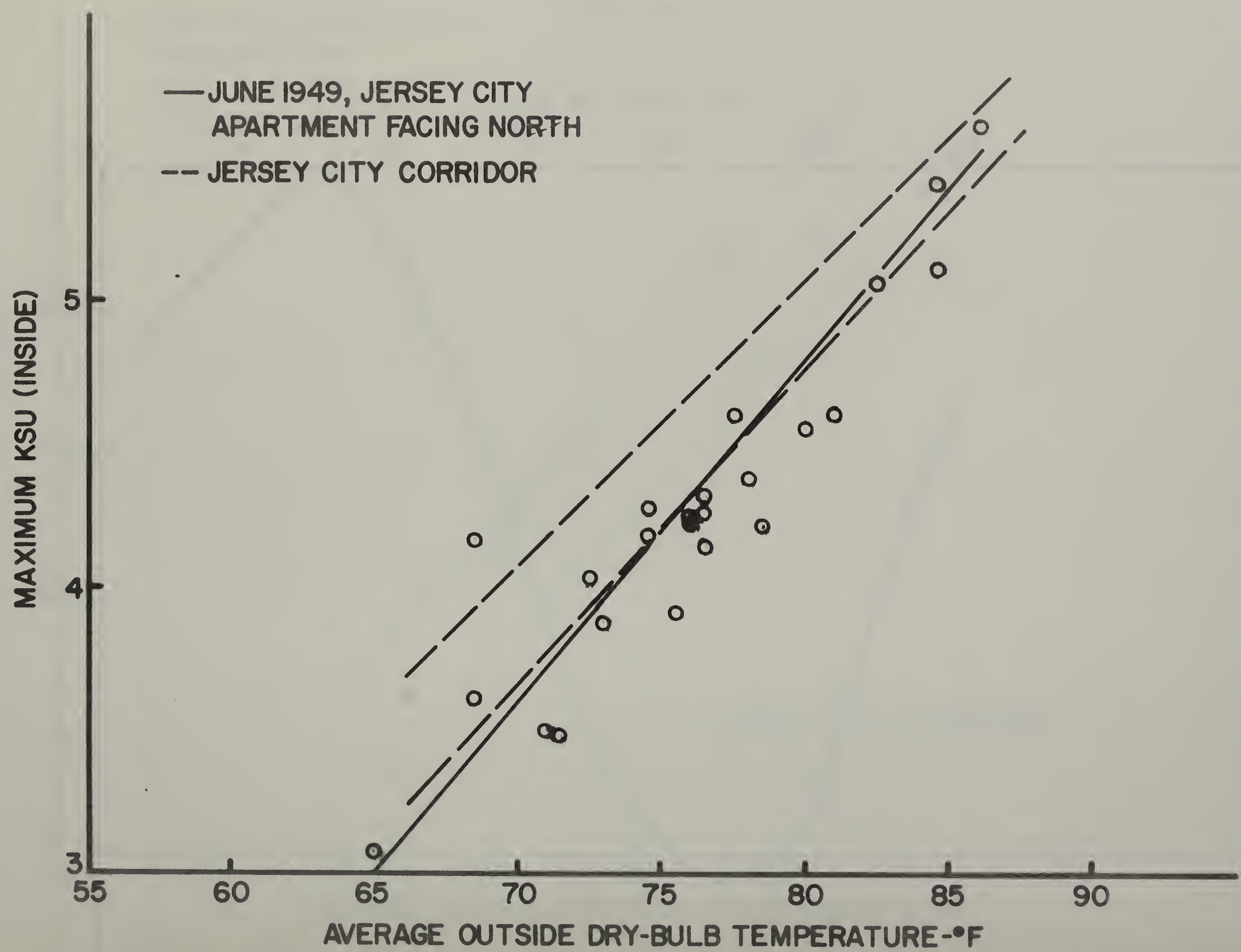
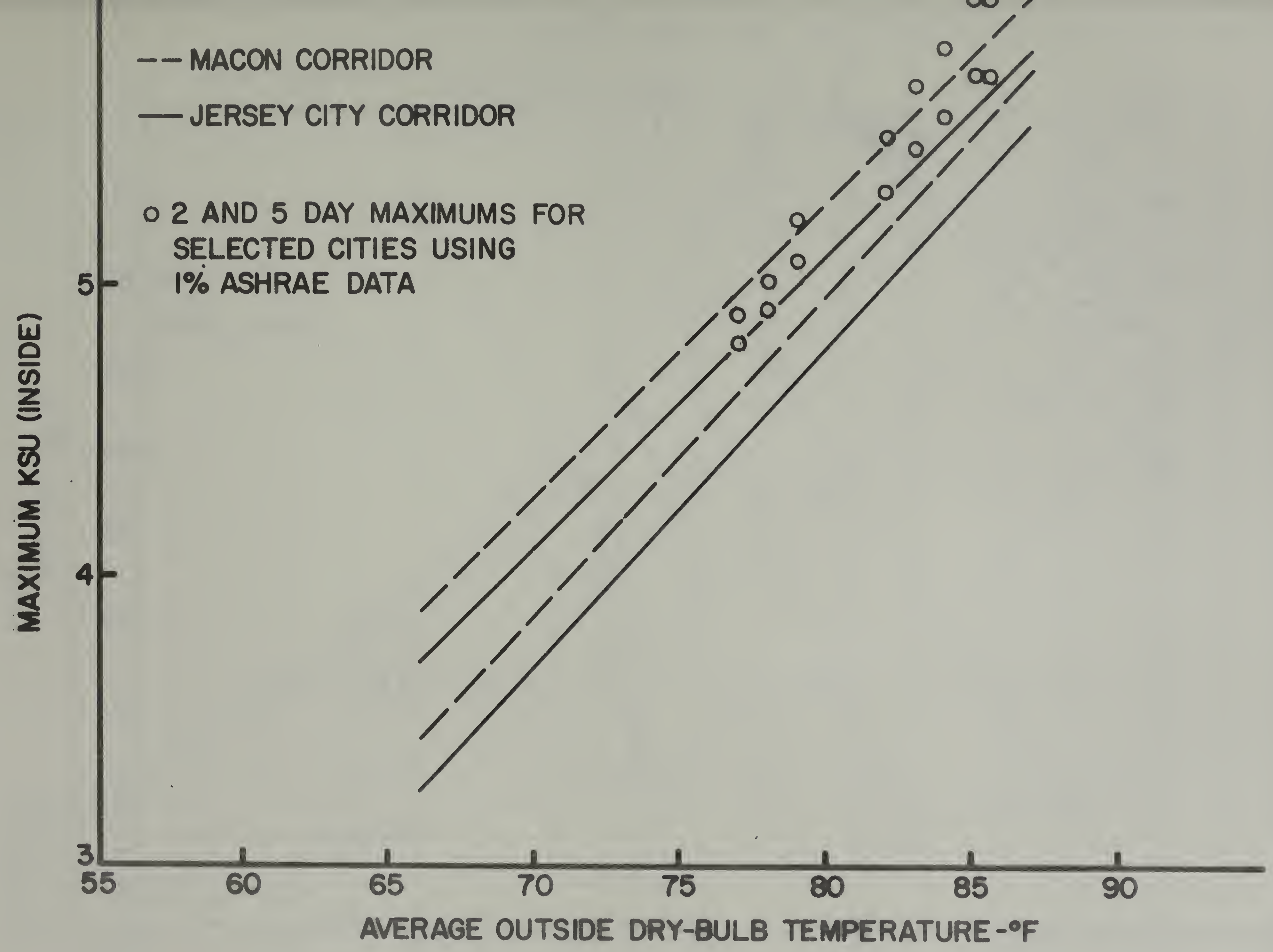
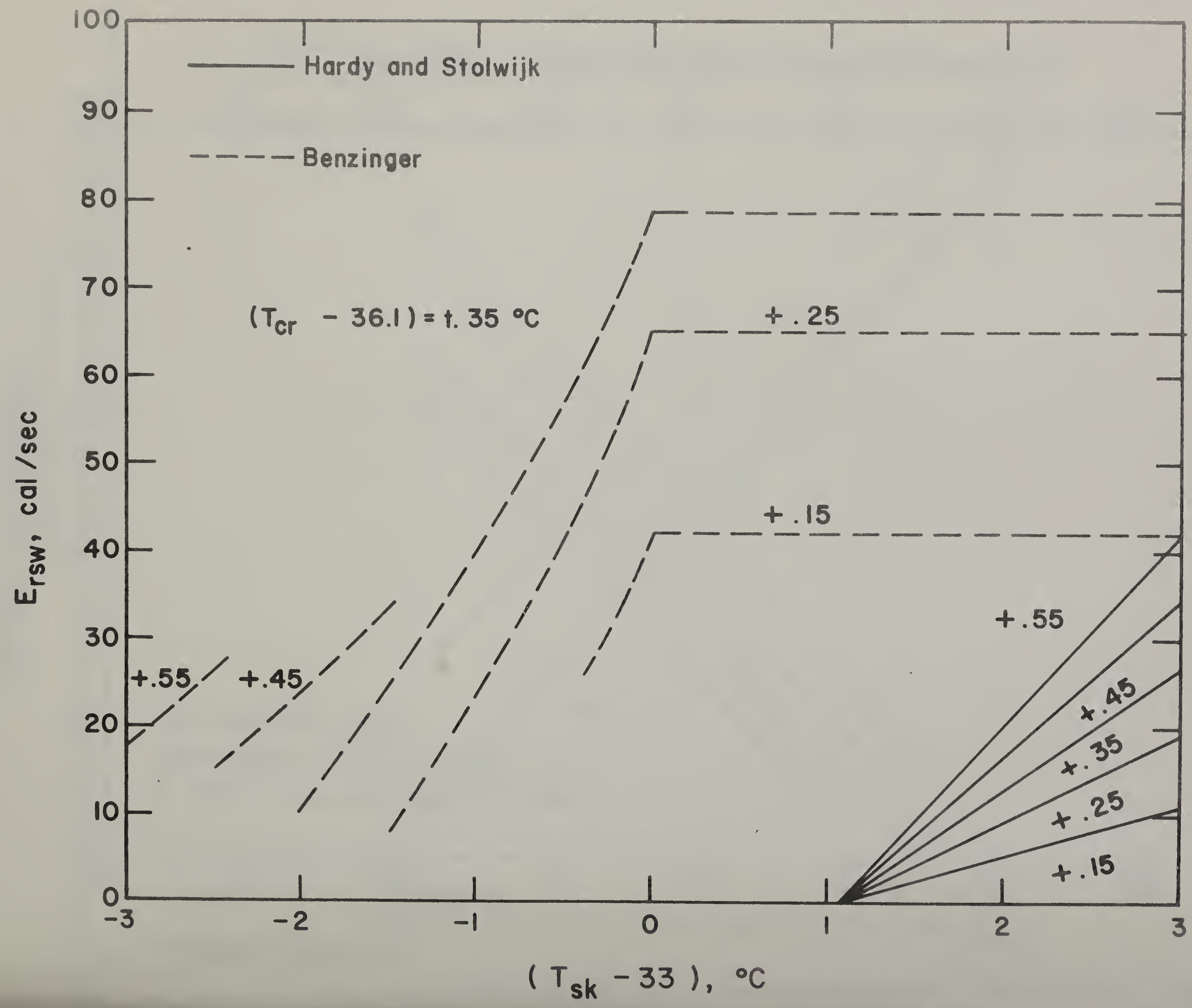

—JUNE 1949, JERSEY CITY
APARTMENT FACING NORTH
-- JERSEY CITY CORRIDOR
MAXIMUM KSU (INSIDE)
3
4
5
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE-°F

Figure 85



-- MACON CORRIDOR
— JERSEY CITY CORRIDOR
o 2 AND 5 DAY MAXIMUMS FOR SELECTED CITIES USING 1% ASHRAE DATA
MAXIMUM KSU (INSIDE)
3
4
5
55
60
65
70
75
80
85
90
AVERAGE OUTSIDE DRY-BULB TEMPERATURE -°F

Figure 87



Hardy and Stolwijk
Benzinger
$(T_{cr} - 36.1) = +.35$ °C
+.25
+.15
+.55
+.45
+.35
+.25
+.15
+.55
+.45
E_{rsw}, cal /sec
$(T_{sk} - 33)$, °C
0
10
20
30
40
50
60
70
80
90
100
−3
−2
−1
0
1
2
3

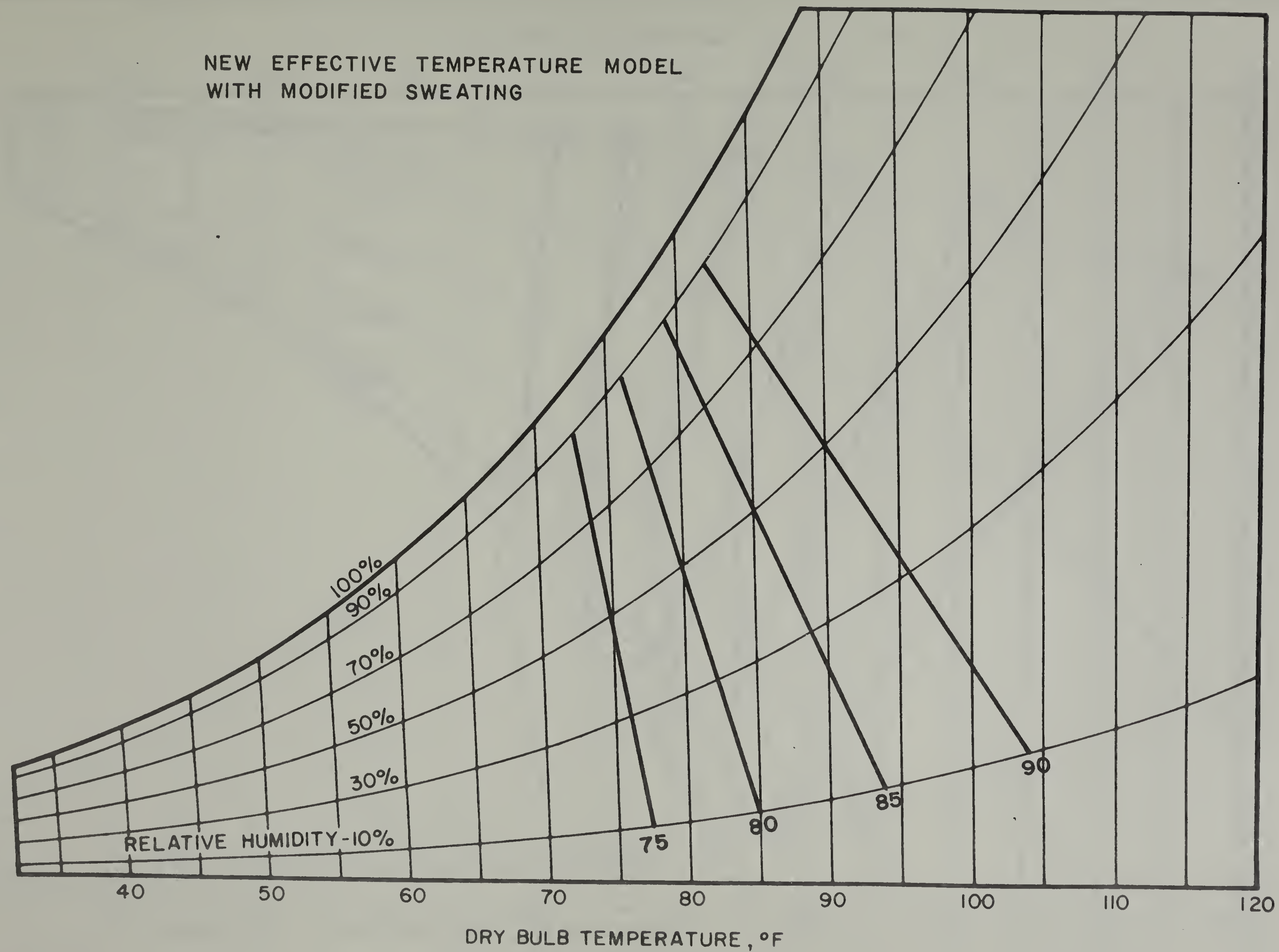

NEW EFFECTIVE TEMPERATURE MODEL
WITH MODIFIED SWEATING
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 89

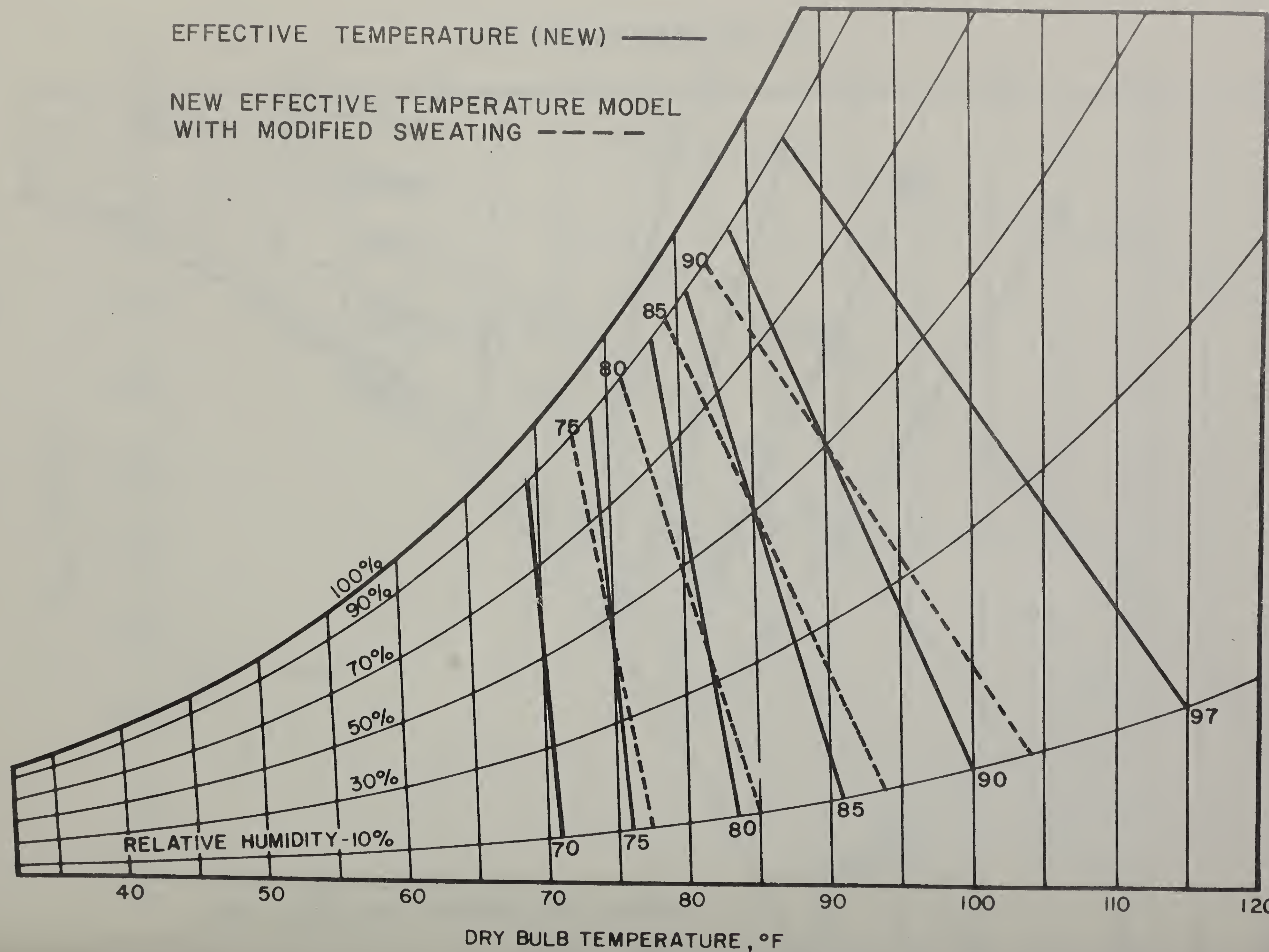

EFFECTIVE TEMPERATURE (NEW) ———
NEW EFFECTIVE TEMPERATURE MODEL
WITH MODIFIED SWEATING - - - -
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
75
80
85
90
70
75
80
85
90
97
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 90

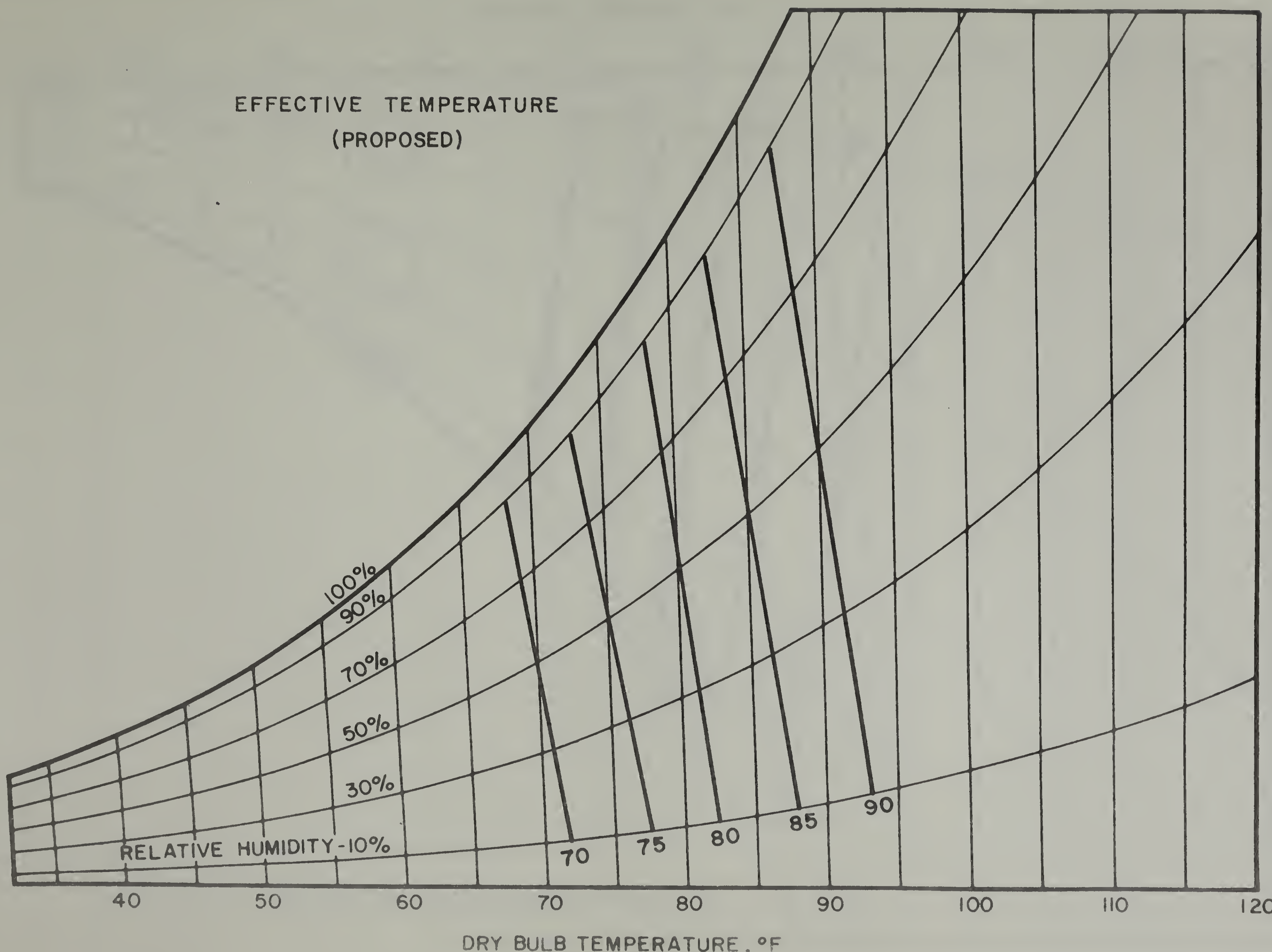

EFFECTIVE TEMPERATURE
(PROPOSED)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 91

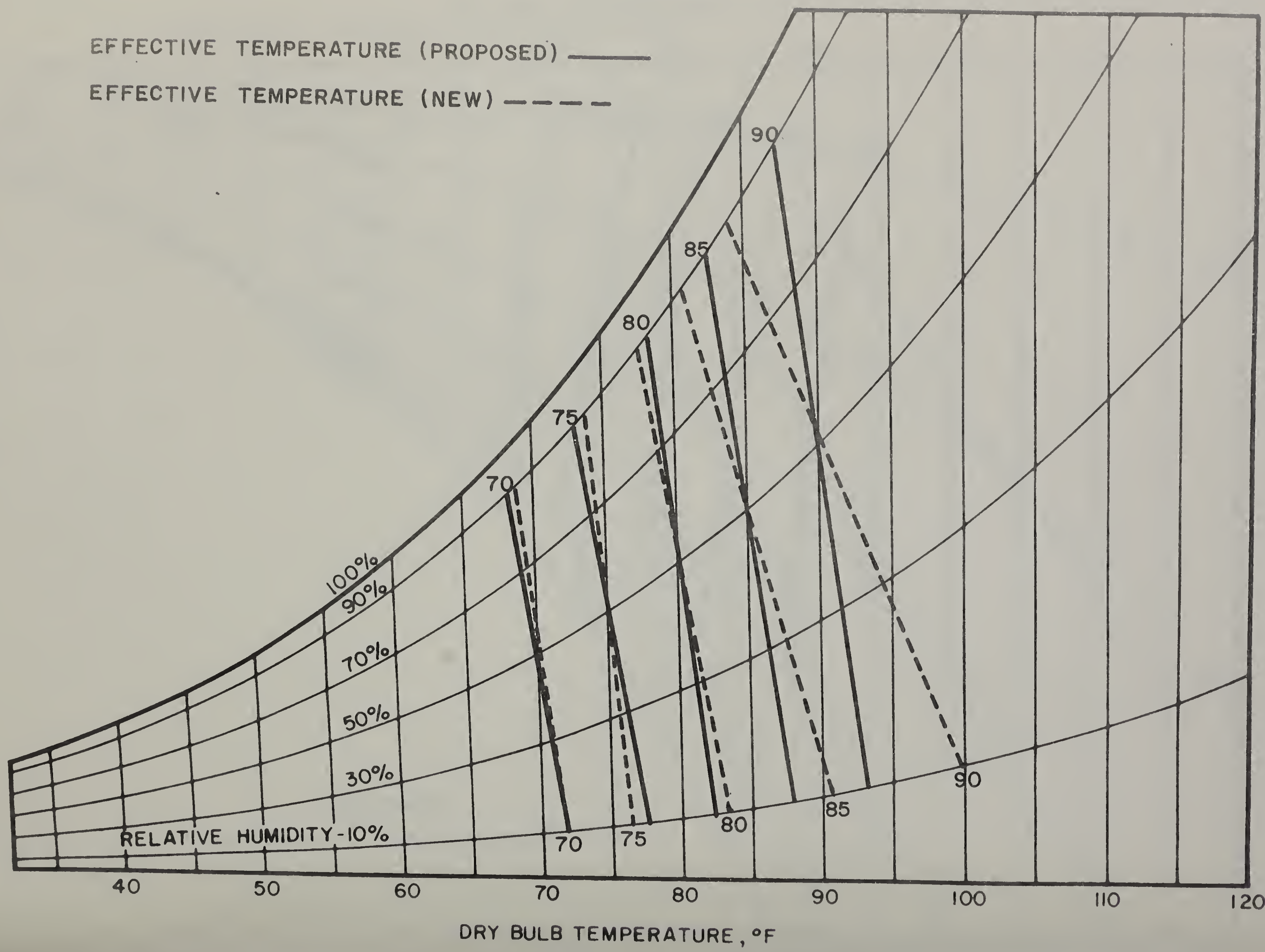

EFFECTIVE TEMPERATURE (PROPOSED)
EFFECTIVE TEMPERATURE (NEW)
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
70
75
80
85
90
70
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 92

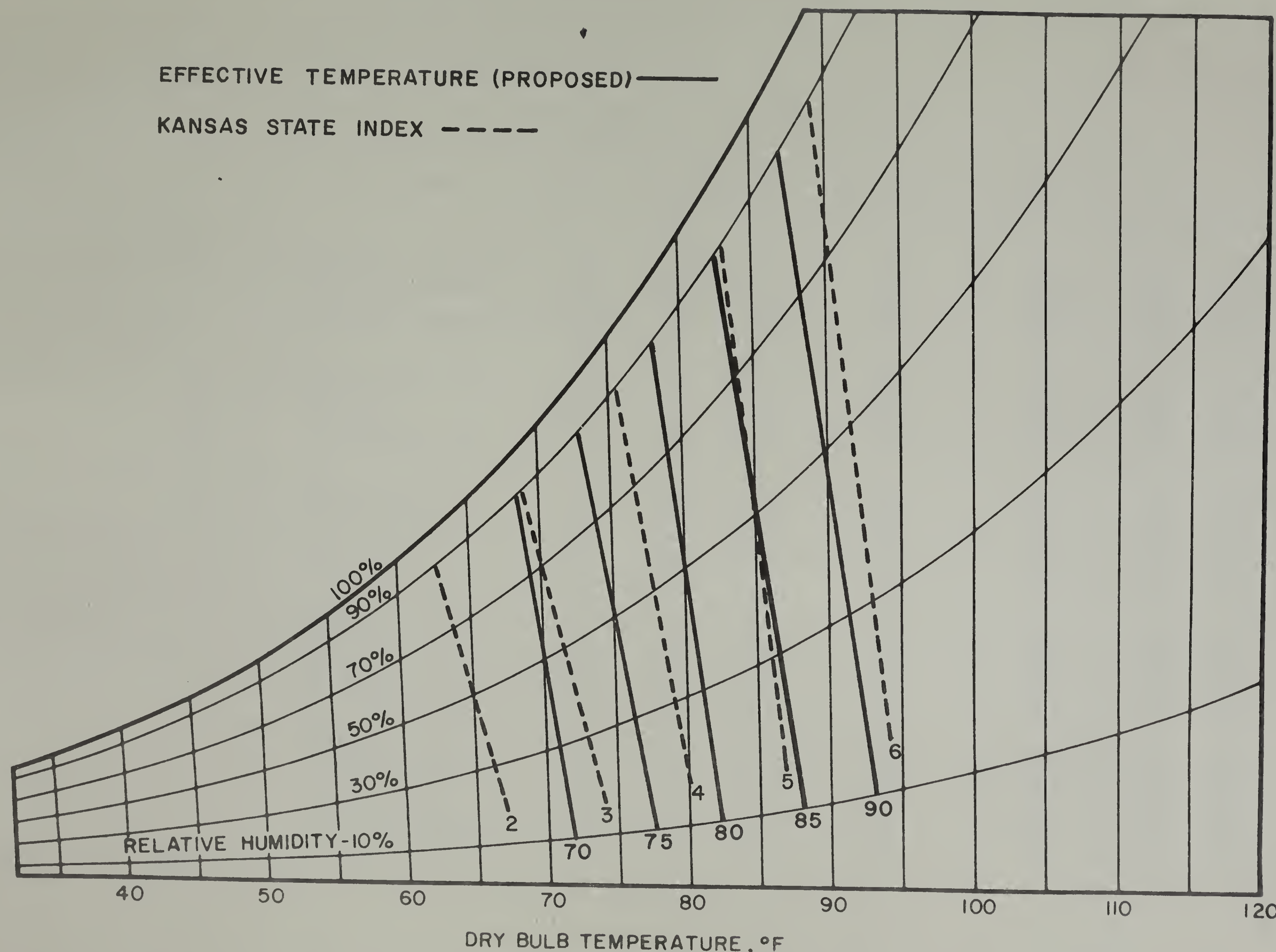

EFFECTIVE TEMPERATURE (PROPOSED)
KANSAS STATE INDEX
100%
90%
70%
50%
30%
RELATIVE HUMIDITY-10%
2
3
4
5
6
70
75
80
85
90
40
50
60
70
80
90
100
110
120
DRY BULB TEMPERATURE, °F

Figure 93

Table 1

HSP ------------ NATIONAL HOMES CORPORATION

HUD FILE NO. --- 1257-01-203

DWELLING TYPE -- SINGLE FAMILY ATTACHED

SITE ----------- INDIANAPOLIS, INDIANA

LAYER	MATERIAL DESCRIPTION	THICKNESS IN.	DENSITY LB/CU.FT.	SP.HEAT BTU/LB-F	TH.CONDUCTIVITY BTU-IN/HR-SQ.FT-F	TH.RESISTANCE HR-SQ.FT-F/BTU	EMISSIVITY OUTSIDE	EMISSIVITY INSIDE
EXTERIOR WALL								
1	AIR FILM(7.5 MPH WIND)	.000	.0	.000	.000	.250	.000	.000
2	WEYERHAEUSER PANEL 15	.312	34.0	.550	.800	.390	.120	.050
3	FIBERGLASS INSULATION	2.000	3.0	.157	.270	7.400	.900	.900
4	AIR SPACE	1.175	.0	.000	.000	1.250	.000	.000
5	FOIL-BACKED GYPSUM BOARD	.500	50.0	.259	1.125	.450	.050	.900
6	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
INTERIOR WALL								
1	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
2	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
3	AIR SPACE	1.500	.0	.000	.000	.970	.000	.000
4	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
5	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
PARTY WALL								
1	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
2	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
3	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
4	F. G. SOUND INSULATION	4.000	3.0	.157	.270	14.800	.900	.900
5	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
6	GYPSUM BOARD	.500	50.0	.259	1.125	.450	.900	.900
7	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
FLOOR/CEILING								
1	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
2	VINYL ASBESTOS TILE	.580	.0	.000	.000	.050	.000	.900
3	PLYWOOD	.750	34.0	.550	.800	.940	.900	.000
4	AIR SPACE	16.500	.0	.000	.000	.970	.000	.000
5	GYPSUM BOARD(CEILING)	.500	50.0	.259	1.125	.450	.900	.900
6	AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
ROOF/CEILING								
1	AIR FILM(7.5 MPH WIND)	.000	.0	.000	.000	.250	.000	.000
2	*240 ASPHALT SHINGLES	.000	70.0	.000	.000	.440	.900	.000
3	2 LAYERS *15 FELT	.000	.0	.000	.000	.120	.000	.000
4	PLYWOOD	.375	34.0	.550	.800	.470	.900	.000
5	ATTIC AIR SPACE	.000	.0	.000	.000	2.800	.000	.000
6	6 IN. F.GL.BATTS W/VB.	6.000	3.0	.157	.310	19.000	.900	.000
7	GYPSUM BOARD(CEILING)	.500	50.0	.259	1.125	.450	.900	.900
8	AIR FILM(STILL AIR)	.000	.0	.000	.680	.000	.000	.000

p

Table 1 - Continued

FIRST FLOOR							
1 AIR FILM(STILL AIR)	.000	.0	.000	.000	.680	.000	.000
2 VINYL ASBESTOS TILE	.058	.0	.000	.000	.050	.900	.900
3 PLYWOOD	.750	34.0	.550	.800	.940	.000	.900
4 CRAWL SPACE, AIR	18.000	.0	.000	.900	.610	.000	.000
5 6 MIL V.B.	.000	.0	.000	.000	.000	.900	.900
BASEMENT WALL							
1 NONE	.000	.0	.000	.000	******	.000	.000
BASEMENT FLOOR							
1 NONE	.000	.0	.000	.000	******	.000	.000
INTERIOR DOOR							
1 H.C. HARDWOOD	1.000	.0	.000	.000	2.270	.900	.900
WINDOW							
1 GLASS (SINGLE GLAZING)	.125	170.0	.200	5.400	.944	.900	.900
EXTERIOR DOOR							
1 S. C. HARDWOOD	1.375	45.0	.550	1.100	2.220	.900	.900

Table 2

HSP ----------- NATIONAL HOMES CORPORATION

HUD FILE NO. --- 1257-01-203

DWELLING TYPE -- SINGLE FAMILY ATTACHED

UNIT TYPE ------ FOUR BEDROOM DUPLEX

SITE ----------- INDIANAPOLIS, INDIANA

FLOOR LEVEL -- 2

TYPE OF ROOM - BEDROOM 1 (LENGTH=11.3 FT. WIDTH=11.3 FT. HEIGHT= 7.5 FT.)

	TOTAL SQ.FT.	E.WALL SQ.FT.	I.WALL SQ.FT.	P.WALL SQ.FT.	OPENING SQ.FT.	WINDOW/DIMENSION SQ.FT.	FT.	SH.FAC.	E.DOOR SQ.FT.	I.DOOR SQ.FT.	FLOOR/CEILING SQ.FT.
S 1	84.	68.	0.	0.	0.	16.	4.00X4.00	.55	0.	0.	0.
S 2	85.	85.	0.	0.	0.	0.	.00X .00	****	0.	0.	0.
S 3	84.	0.	84.	0.	0.	0.	.00X .00	****	0.	0.	0.
S 4	85.	0.	68.	0.	0.	0.	.00X .00	****	0.	17.	0.
TOTAL		153.	153.	0.	0.	16.			0.	17.	128.

TOTAL INTERIOR DOOR CRACK- 18.3 FT.
TOTAL EXTERIOR DOOR CRACK- .0 FT.
TOTAL WINDOW CRACK- 16.0 FT.
ROOF AREA- 130.7 SQ. FT.
LIGHTING= .0 WATTS
FURNITURE= .000

FLOOR LEVEL -- 2

TYPE OF ROOM - BEDROOM 2 (LENGTH= 8.0 FT. WIDTH=11.3 FT. HEIGHT= 7.5 FT.)

	TOTAL SQ.FT.	E.WALL SQ.FT.	I.WALL SQ.FT.	P.WALL SQ.FT.	OPENING SQ.FT.	WINDOW/DIMENSION SQ.FT.	FT.	SH.FAC.	E.DOOR SQ.FT.	I.DOOR SQ.FT.	FLOOR/CEILING SQ.FT.
S 1	60.	44.	0.	0.	0.	16.	4.00X4.00	.55	0.	0.	0.
S 2	85.	0.	68.	0.	0.	0.	.00X .00	****	0.	17.	0.
S 3	60.	0.	60.	0.	0.	0.	.00X .00	****	0.	0.	0.
S 4	85.	0.	85.	0.	0.	0.	.00X .00	****	0.	0.	0.
TOTAL		44.	213.	0.	0.	16.			0.	17.	91.

TOTAL INTERIOR DOOR CRACK- 16.7 FT.
TOTAL EXTERIOR DOOR CRACK- .0 FT.
TOTAL WINDOW CRACK- 16.0 FT.
ROOF AREA- 91.0 SQ. FT.
LIGHTING= .0 WATTS
FURNITURE= .000

p

Table 3

HSP ----------- NATIONAL HOMES CORPORATION

HUD FILE NO. --- 1257-01-203

DWELLING TYPE -- SINGLE FAMILY ATTACHED

UNIT TYPE ------ FOUR BEDROOM DUPLEX

SITE ----------- INDIANAPOLIS, INDIANA

	E.WALL	I.WALL	P.WALL	FL./CEIL.	ROOF	1ST FL.	B.FLOOR	B.WALL	I.DOOR	WINDOW	E.DOOR
TOTAL AREA/UNIT (SQ.FT.)	915.	1731.	317.	1065.	533.	533.	0.	0.	200.	143.	32.
WT./UNIT AREA(LB./SQ.FT.)	3.467	4.167	9.333	4.208	4.646	2.125	.000	.000	.000	1.771	5.156
U-VALUE(BTU/HR-SQ.FT.-F)	.096	.310	.056	.265	.042	.439	.000	.000	.441	1.059	.450
TH.MASS(BTU/SQ.FT.-F)	1.104	1.079	2.315	1.708	1.359	1.169	.000	.000	.000	.354	2.836
TH.TIME CONST.(HR.)	5.691	1.743	20.792	2.948	16.384	1.402	.000	.000	.000	.167	3.148
SOLAR ABSORPTIVITY	.280	****	****	****	.930	****	****	****	****	.000	.850
SOLAR TRANSMITTANCE	****	****	****	****	****	****	****	****	****	.870	****

TOTAL FLOOR AREA PER UNIT = 1065. SQ.FT.
AREA RATIO, EXT.WALL/FLOOR = .859
AREA RATIO, WINDOW/EXT.WALL = .157

Table 4 Evaluation of Index of Heat Stress

Index of Heat Stress	Physiological and Hygienic Implications of 8-Hr Exposures to Various Heat Stresses
0	No thermal strain
+10	Mild to moderate heat strain. Where a job involves higher intellectual functions, dexterity, or alertness, subtle to substantial decrements in performance may be expected. In performance of heavy physical work, little decrement expected unless ability of individuals to perform such work under no thermal stress is marginal.
40 50 60	Severe heat strain, involving a threat to health unless men are physically fit. Break-in period required for men not previously acclimatized. Some decrement in performance of physical work is to be expected. Medical selection of personnel desirable because these conditions are unsuitable for those with cardiovascular or respiratory impairment or with chronic dermatitis. These working conditions are also unsuitable for activities requiring sustained mental effort.
70 80 90	Very severe heat strain. Only a small percentage of the population may be expected to qualify for this work. Personnel should be selected (a) by medical examination and (b) by trial on the job (after acclimatization). Special measures are needed to assure adequate water and salt intake. Amelioration of working conditions by any feasible means is highly desirable, and may be expected to decrease the health hazard while increasing efficiency on the job. Slight "indisposition" which in most jobs would be insufficient to affect performance may render workers unfit for this exposure.
100	The maximum strain tolerated daily by fit, acclimatized young men.

Table 5 Thermal Performance of Camci Apartment in Jersey City, New Jersey

Month and Year	Total No. of Hours in 4 Months $T_{dry\text{-}bulb} \geq 80$ °F	Total No. of Hours in 4 Months $T_{wet\text{-}bulb} \geq 67$ °F	Percent of Time New ET ≥ 81	Percent of Time New ET ≥ 85	Percent of Time KSU ≥ 5	Percent of Time PMV ≥ 1 (0.25 clo)	Percent of Time HSI ≥ 40
1949	613	1397	-	-	-	-	-
June	-	-	-	11.0	8.1	7.9	0.0
July	-	-	-	27.6	22.4	20.6	0.5
August	-	-	-	11.7	8.7	8.2	0.0
September	-	-	-	0.0	0.0	0.0	0.0
1950	506	822	-	-	-	-	-
June	-	-	11.7	0.8	0.6	0.7	0.0
July	-	-	20.3	3.2	2.0	1.9	0.0
August	-	-	9.9	3.1	1.5	0.7	0.0
September	-	-	5.8	1.1	0.4	0.1	0.0
1951	-	1086	-	-	-	-	-
June	-	-	9.2	1.0	0.6	0.3	0.0
July	-	-	30.6	5.5	2.0	1.6	0.0
August	-	-	22.6	3.5	0.5	0.4	0.0
September	-	-	0.8	0.0	0.0	0.0	0.0
1952	-	1258	-	-	-	-	-
June	-	-	18.9	8.7	6.1	6.0	1.1
July	-	-	60.8	27.0	21.1	19.4	0.3
August	-	-	20.6	1.1	0.4	0.1	0.0
September	-	-	3.3	0.1	0.0	0.0	0.0
1953	612	1069	-	-	-	-	-
June	-	-	24.3	5.7	4.0	3.6	0.0
July	-	-	42.5	16.5	11.7	10.8	0.0
August	-	-	20.8	8.5	7.5	7.1	0.0
September	-	-	17.9	13.9	12.2	11.1	0.0
1954	493	583	-	-	-	-	-
June	-	-	8.7	1.0	1.0	1.1	0.0
July	-	-	22.8	4.7	4.2	4.3	0.0
August	-	-	8.1	3.1	2.4	2.6	0.0
September	-	-	3.1	0.0	0.0	0.0	0.0
1955	616	1292	-	-	-	-	-
June	-	-	4.7	0.0	0.0	0.0	0.0
July	-	-	70.8	35.9	26.1	25.5	0.0
August	-	-	51.2	24.9	17.3	15.9	0.0
September	-	-	0.0	0.0	0.0	0.0	0.0
1956	390	887	-	-	-	-	-
June	-	-	17.9	5.1	3.9	3.9	0.0
July	-	-	10.5	1.7	0.8	0.7	0.0
August	-	-	21.5	3.4	1.9	1.3	0.0
September	-	-	2.4	0.0	0.0	0.0	0.0
1957	617	1123	-	-	-	-	-
June	-	-	36.2	13.3	9.4	8.5	0.0
July	-	-	34.9	12.4	10.5	10.9	0.0
August	-	-	17.5	4.6	3.2	3.5	0.0
September	-	-	11.9	0.7	0.4	0.4	0.0
1958	463	1074	-	-	-	-	-
June	-	-	1.0	0.0	0.0	0.0	0.0
July	-	-	43.3	13.2	7.0	6.5	0.0
August	-	-	28.1	3.5	1.2	0.9	0.0
September	-	-	3.7	0.0	0.0	0.0	0.0

Table 6 ASHRAE 1% Design Weather Conditions for Selected Cities

Location	T_{db}, °F	Range, °F	$T_{db} - \frac{Range}{2}$
Portland, Maine	88.0	22.0	77.0
Concord, N. H.	91.0	26.0	78.0
Hartford, Conn.	90.0	22.0	79.0
West Chester, Pa.	92.0	20.0	82.0
New York, N. Y. (Kennedy)	91.0	16.0	83.0
Jersey City, N. J.	94.0	20.0	84.0
Washington, D. C.	94.0	18.0	85.0
New York, N. Y. (Central Park)	94.0	17.0	85.5

Table 7 Range of Application of Various Physiological Indices as Calculated in This Report

Index	Metabolic Production	Clothing	Dry-bulb Temperature	Wet-bulb Temperature or Relative Humidity	Air Velocity
ET	rest only	Summer clothing	1 - 43 °C	1 - 43 °C	.1 - 3.5 m/sec
RT	rest only	Light clothing	18 - 45 °C	18 - 45 °C	.1 - 3.0 m/sec
KSU	rest only	0.6 clo	60 - 98 °F	15% - 85%	< 45 fpm
PMV	50 $\frac{kcal}{m^2\ hr}$	0.25 clo and 0.60 clo	-	-	.1 m/sec
ET (new)	58.2 $\frac{watts}{m^2}$	0.60 clo	-	-	.1 m/sec
HSI (original)	100 - 500 kcal/hr	unspecified	27 - 60 °C	15 - 35 °C	.25 - 10.0 m/sec
HSI	58.2 $\frac{watts}{m^2}$	0.60 clo	-	-	.1 m/sec
Thermal Comfort when Equil. is Maintained by Sweating	58.2 $\frac{watts}{m^2}$	0.60 clo	-	-	.1 m/sec
ITS	100 - 600 kcal/hr	Military overalls over shorts	20 - 55 °C	15 - 35 °C	.1 - 3.5 m/sec
P4SR	100 - 350 kcal/hr	Wearing only shorts	27 - 55 °C	15 - 36 °C	.05 - 2.5 m/sec